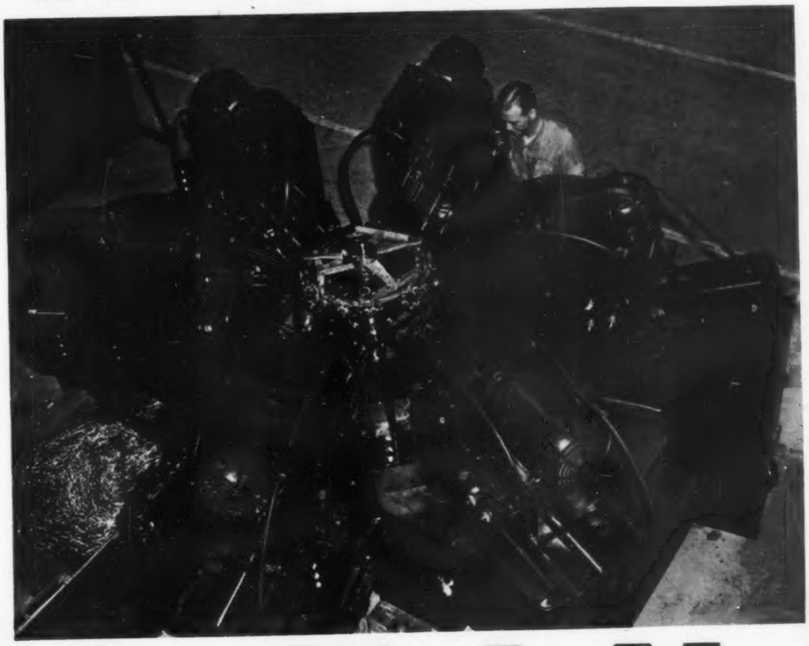
August 1941

MACHINE



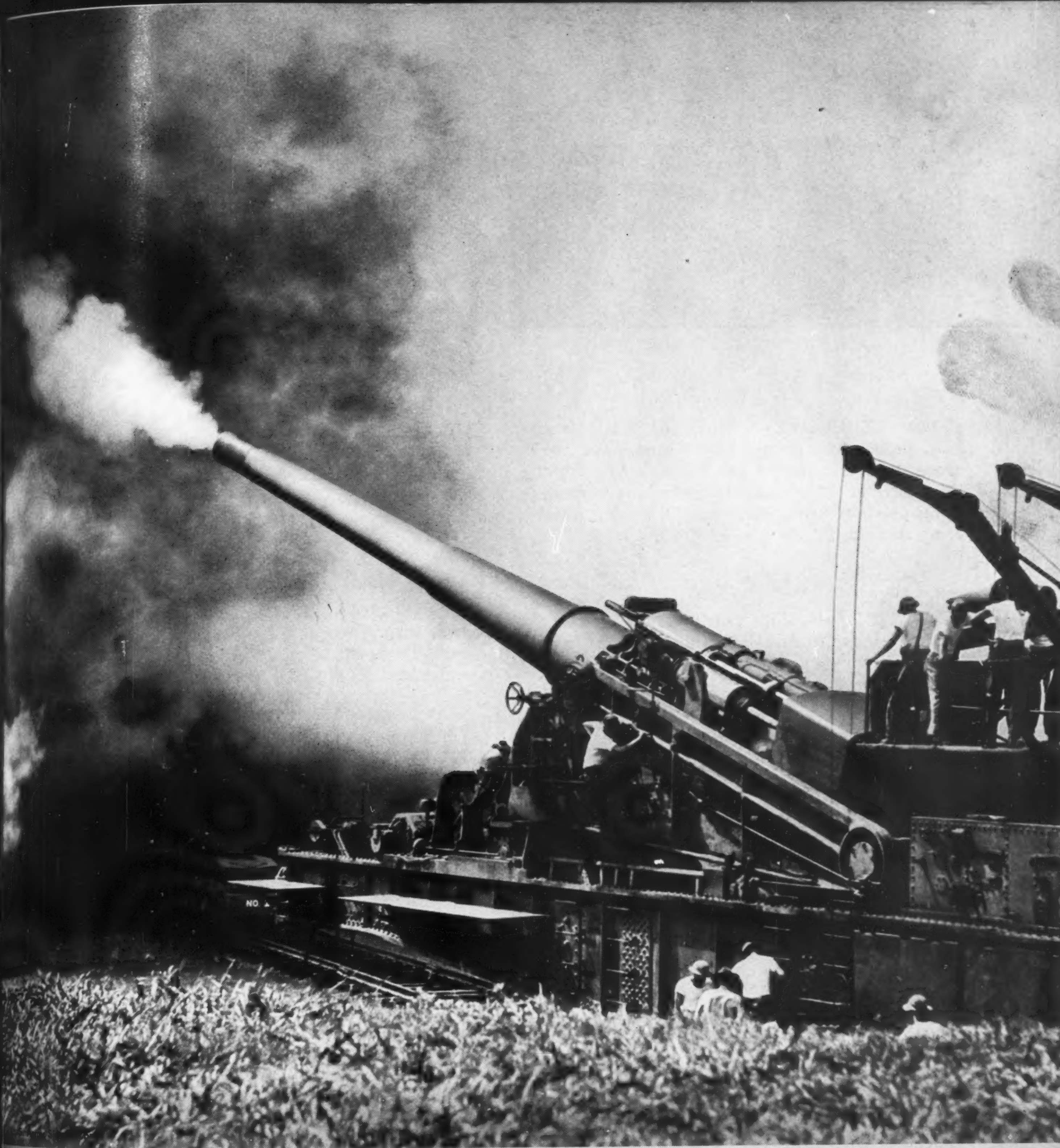
DESIGN

In This Issue:

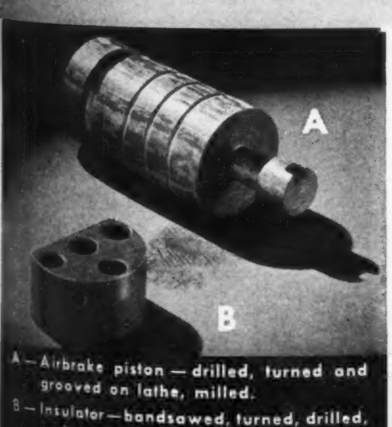
Defense Program Poses Serious Problems

Calculating Stresses in Shaft Design

Aircraft Uses of Antifriction Bearings



RADIO CALLS ITS SHOTS



A — Airbrake piston — drilled, turned and grooved on lathe, milled.
B — Insulator — bandsawed, turned, drilled, and counterbored.

Army and Signal Corps engineers see to it that there are no weak links in the chain of communications controlling gunfire or troop movement. • Bakelite-laminated insulates many important circuits, here as in other less spectacular, but no less essential, operations. SYNTHANE CORPORATION, OAKS, PENNA.

SYNTHANE TECHNICAL PLASTICS

SHEETS • RODS • TUBES • FABRICATED PARTS

SYNTHANE
Bakelite-laminated

SILENT STABILIZED GEAR MATERIAL

Topics

COMPREHENSIVE revisions of S.A.E. steel specifications provide for 72 carbon and alloy grades and 12 corrosion and heat-resisting alloys. This total of 84 replaces the former 109 grades in the previous series. Such simplification is a welcome step during the present defense crisis.

♦

INVENTORS are urged to apply their knowledge and experience to solving national defense problems, according to an appeal of the National Inventors' Council of the Department of Commerce. It is suggested that each head of research and development in private industry submit to the council those ideas appearing to have some interest for the Army and Navy.

♦

ROXAPRENE—a corrosion-resistant, fast-setting synthetic applied by dip, spray or roller —is unaffected by 2 per cent caustic solution after 600 hours immersion. It also is fairly resistant to dilute hydrochloric acid solutions. This coating when used instead of galvanizing releases a strategic material, zinc, for other defense purposes.

♦

PRIZES to be awarded by the American Welding Society for papers which advance the art of welding aircraft steels have been set up by the Summerill Tubing Co., Bridgeport, Pa. The papers may treat any type of welding which is or can be used for the fabrication of structures or assemblies in the production of aircraft steels. They may cover any phase of the problem whether concerning design, fabrication or laboratory investigations.

♦

NEW facilities for producing plastic compounds are needed to relieve the present shortage of basic chemicals such as ammonia and methyl alcohol. Otherwise we soon will be faced with an acute shortage of plastics.

TRACKLESS tanks utilizing eight wheels have been proposed as the answer to Germany's "mastodons". Although considered expensive—$35,000 each—they will be given a trial by the army.

♦

DUPONT is enlarging plant facilities for producing 19,000 tons of neoprene annually to provide a supply adequate for all defense and commercial needs including a substantial tonnage for the manufacture of heavy duty tires.

♦

MORE than 1,500,000 pounds of aluminum have been released by one electric appliance manufacturer by utilizing plastics, steel and other substitute materials. Sufficient to provide for building 130 two-engine army bombing planes, this substitution is an indication of the resourcefulness of American engineers and industry. Further, the company has found that in many cases better materials and methods have been developed than formerly were used.

♦

WATER is now being treated with synthetic resins to effect a new method of complete water purification. Two types of resins are used—polyhydric phenol formaldehyde for changing calcium chloride to hydrochloric acid and amine formaldehyde to remove the acid itself.

♦

IN DOING work for British war machines, American designers frequently are called upon to meet specifications considerably different from those encountered in American practice. A curious example occurred recently. The helix angle for helical gears in the transmission of a British army tank was specified as 29 degrees 53 minutes. But the odd angle puzzled the designer working on the job, particularly since there was only 7 minutes difference between the specified angle and the conventional 30-degree helix angle.

Specifying Materials in Design of Pressure Vessels

By A. Grodner
Blaw-Knox Co.

HARDLY an industry exists which does not utilize some form of a pressure vessel. It may be only an air tank, a gas storage container or a hydraulic accumulator, used as an auxiliary unit to a machine, or it may be the final processing vessel in a chemical plant. The proper design of such pressure vessels therefore claims the attention of many design engineers.

Since the advent of high pressures and temperatures it is imperative that the design engineer be familiar with, and keep abreast of, the latest data on materials of construction, on design stresses and on practicable, economical shop fabricating methods. The A. S. M. E. code crystallizes the seasoned and the tried ideas into a most valuable and workable handbook of pressure vessel design. In many of our states and in Canada, the A. S. M. E. code is mandatory for all pressure vessels; in the other states the code is the recognized unofficial authority.

In the design of a pressure vessel our first consideration is the choice of the materials of con-

Fig. 1 — Elliptical heads used on 120-gallon, steam-jacketed autoclave are capable of operating under working pressures of 2000 pounds per square inch

struction. Such a choice must be governed by:

a. Adequacy of the material to withstand the maximum pressure and temperature which may be encountered in the process
b. Adequacy of the material to resist corrosive agents or reactions where such conditions prevail
c. Ability to fabricate the material in forming it into various shapes, in welding, in machining, in finishing, etc.
d. Relative costs of the several available materials.

The following summary will be useful in arriving at a possible solution to this problem. It will not, by any means, embrace all available materials, but will serve as a nucleus and as a guide.

Pressure Vessel Materials Summarized

1. The most widely used steel for general purposes in the construction of pressure vessels is that complying with A. S. T. M. specification A-70. This steel has an ultimate tensile strength of 55,000 pounds per square inch through a range of metal temperatures of from —20 to 650 degrees Fahr. with a yield point at approximately one-half its tensile strength. Besides possessing good ductility, it forms, welds and machines easily. It also is the least expensive of the steels suitable for pressure vessels

2. For higher pressures, or for large diameter vessels, the thickness of the wall and consequently the weight of the vessel can be reduced by the use of a higher tensile strength steel, complying with A. S. T. M. A-212. This steel has a tensile strength of 70,000 pounds per square inch, so that the vessel wall is reduced in direct proportion to 55/70, or to about 79 per cent. Although the pound price of this steel is higher than that of A. S. T. M. A-70, the overall cost of the vessel is in most cases lower and its lighter weight may even permit lighter foundations. This steel also lends itself easily to fabrication

3. At elevated temperatures (over 650 degrees Fahr. metal temperature), most steels rapidly lose tensile strength and, even more seriously, decline in creep strength. For such conditions, a steel complying with A. S. T. M. specification A-204, containing .4 to .6 per cent molybdenum, has proved satisfactory, not only because it retains a safe tensile strength but also a very good degree of creep strength. The graph in *Fig.* 3 shows clearly the relative tensile strengths of the three steels at higher temperatures, molybdenum being strongest.

4. At sub-zero temperatures most steels gain in tensile strength but lose rapidly in ductility, making them subject to failure through sudden impacts. A satisfactory steel for such temperature conditions must therefore retain not only high tensile strength but continued ductility through the range of temperature. The A. S. M. E. code requires the material to register, at the lowest temperature encountered, a Charpy impact value of 15 foot pounds on a 10 x 10 millimeter keyhole notch impact specimen. For temperatures down to about —50 degrees Fahr. a close grained normalized carbon steel (modified A. S. T. M. A-70) is adequate and meets the A. S. M. E. requirements. This material is, of course, the least expensive of the available steels for these service conditions. For still lower temperatures, down to —150 degrees Fahr., it is necessary to use nickel steels. One such material, having 2 to 2¾ per cent nickel, corresponding to A. S. T. M. specification A-203, is satisfactory. On the basis of carbon content, three grades are available with minimum tensile strengths of 65, 70 and 75,000 pounds per square inch. The 18-8 chrome nickel grades of stainless steel also possess excellent impact values at sub-zero temperatures, but their cost is much higher than the others

5. For corrosive service conditions where the stainless steel must be used in the construction of a pressure vessel, the A. S. M. E. code now recognizes the types in TABLE I

Columbium Facilitates Fabrication

Tensile strength of 75,000 pounds per square inch for these grades of stainless steel is given in the code. The most widely used and the more easily fabricated grades are types 304 and 347. The latter, "stabilized" with columbium, has been the most effective in reducing carbide precipitations along the grain boundaries, to a minimum. In the early days of welding of the stainless steels such carbides were precipitated at the welding temperatures, with a decided diminution in the corrosion resistance of the material

The straight chrome steels (without nickel) have their important applications also, but greater care and skill are required for their fabrication. They are widely used to resist the action of nitric acid. Their base price is considerably lower than those of the 18-8 stainless, although their fabrication

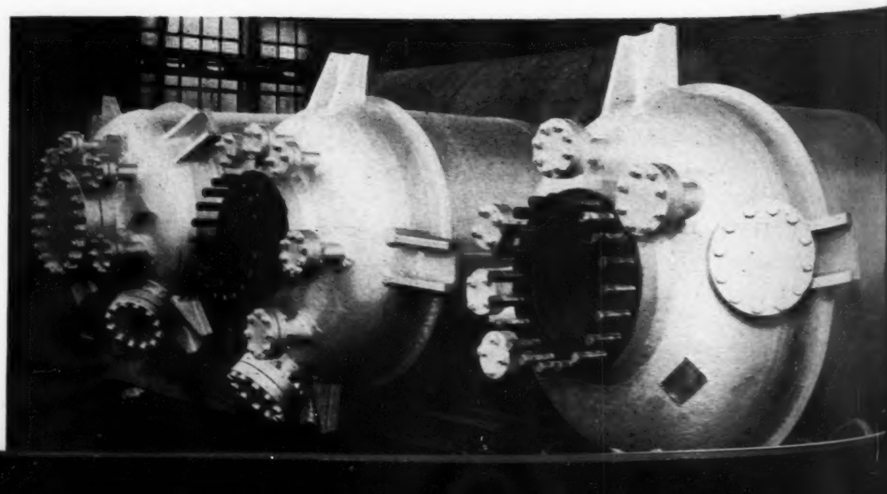


Fig. 2—Three all-welded and X-rayed steel autoclaves are designed for 750 pounds per square inch working pressure at 800 degrees Fahr.

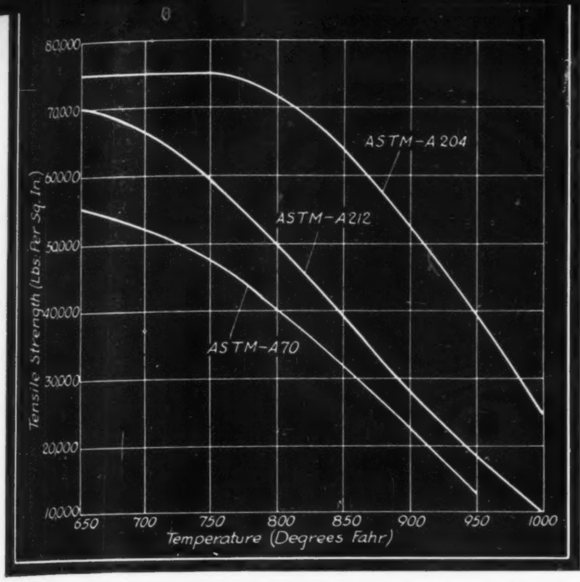


Fig. 3—At elevated temperatures the tensile and creep strength of molybdenum-containing steels materially exceeds that of carbon steels

costs are somewhat higher. Thus far they are not recognized by the A. S. M. E. code, but their adoption is contemplated in the near future. With rigid priorities on nickel at the present time the use of the straight chromes will be more extensive during the coming months

6. All of these mentioned stainless steels can be obtained in the clad form as well as in the solid, with the cladding usually 20 per cent of the total plate thickness. The A. S. M. E. code has just granted permission to utilize the composite thickness of clad plates in design computations (formerly only the steel base material could so be treated in the determination of the wall thickness). The allowable tensile strength of the composite plate is that corresponding to the base material. Obviously, the chief advantage of clad plates is in its lower pound price (No. 304—18 cents; No. 347—21 cents:

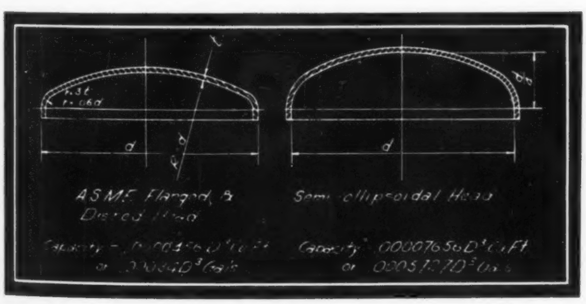


Fig. 4—Flanged and dished head shown on left must be considerably thicker than the shell. Semi-ellipsoidal head is for moderate pressures only

No. 316—24 cents), but since the clad plate must be 75/55 times as thick as the solid stainless plate, or approximately 36 per cent heavier, the total overall cost of a pressure vessel may not be greatly reduced in changing from a solid to a clad plate

7. Among the nonferrous materials sanctioned by the A. S. M. E. code may be listed:

a. Monel (approximately 65 per cent nickel, 35 per cent copper), with tensile strength of 70,000 pounds per square inch at room temperature up to 400 degrees Fahr. dropping down to 62,500 pounds per square inch at 550 degrees Fahr.

b. Copper-silicon alloy, known commercially as "Everdur" or "Herculoy" (approximately 95 per cent copper, 3 per cent silicon) with tensile strength of 50,000 pounds per square inch from room temperature up to 250 degrees Fahr. and dropping to 25,000 pounds per square inch at 350 degrees Fahr.

c. Copper, with tensile strength of 30,000 pounds per square inch at room temperature and up to 150 degrees Fahr. dropping to 20,000 pounds per square inch at 400 degrees Fahr.

d. Nickel. This material is not yet officially recognized by the A. S. M. E. code, and it can therefore be utilized for a pressure vessel only as a clad plate (from 10 to 20 per cent of the total plate thickness). The tensile strength and thickness of the base plate only are used in computations of wall thickness.

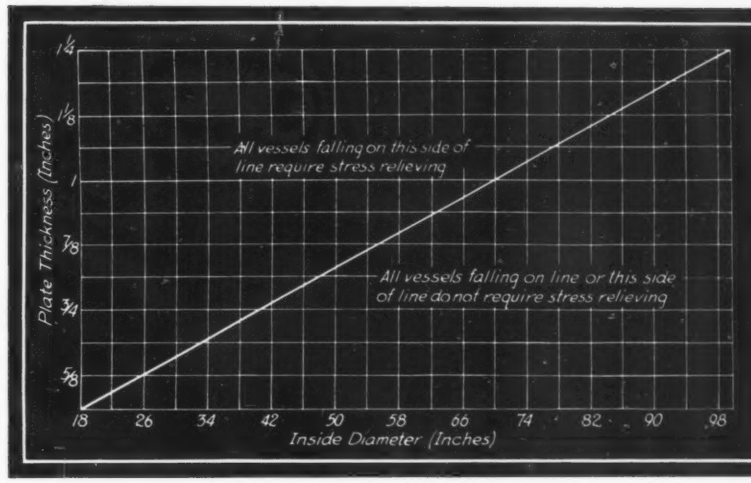


Fig. 5—Necessity for stress relieving in compliance with paragraph U-69 of the Code depends upon the relation between plate thickness and inside diameter

All unfired pressure vessels are subject to the rules of the A. S. M. E. code when both the product of (P—15) (D—4) is greater than 60, and (P—15) (V—1.5) is greater than 22.5, where P = internal working pressure in pounds per square inch, D = inside diameter of vessel in inches and V = volume in cubic feet.

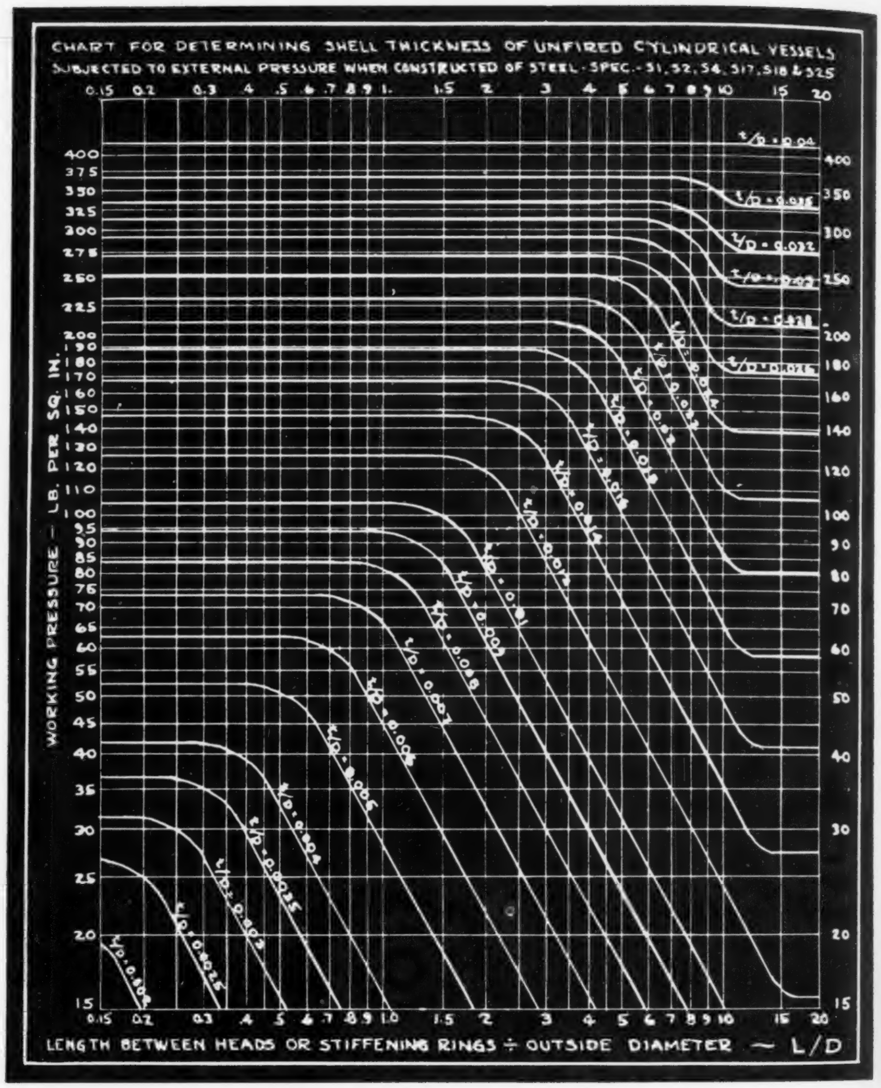
When the interior chamber of a jacketed vessel is open and not subject to pressure, the factor V applies only to the jacket. When both the interior chamber and the jacket are subject to pressure, the factor V applies to the entire volume of the vessel

TABLE I

Stainless Steels for Corrosive Service

Type No.	Chrome (% Min.)	Nickel (% Min.)	Stabilizing Element	Base Price (Cents/lb.)
304	18	8	....	29
316	17	9.5	Molybdenum 2% Min.	54
321	17	9.5	Titanium 5 x Carbon	34
347	17	9.5	Columbium 10 x Carbon	38

Fig. 6—Chart facilitates calculation of shell thicknesses of unfired vessels subject to external pressure



including the jacket.

Three classifications govern the design and construction of fusion welded pressure vessels. The highest quality of welding is that designated by paragraph U-68 of the code. To comply with the rules of this paragraph the fabricator must:

a. X-ray all of the longitudinal and all of the circumferential butt welds (up to a maximum of 5¼-inch thick wall). A prerequisite of the X-ray investigation of the welds is the grinding of the welds inside and outside for uniformity in comparing the results of the exposures
b. Stress relieve the entire vessel in an annealing furnace (for carbon steels, at a temperature of 1150 degrees Fahr. for a period of time equal to one hour per inch of thickness of the vessel wall) to relieve local stresses induced by the heat of welding.
c. Prepare test plates, using the same technique and same material as for the tank proper. This test plate is checked for tensile strength, ductility and density of weld. All welding must be performed by welders who have demonstrated their ability by definite prescribed tests given periodically
d. Subject the completed vessel to a hydrostatic test of double the maximum working pressure, after having had its welds sharply hammered while it is under hydrostatic pressure of one and one-half times the working pressure.

A vessel so constructed is recognized as having an efficiency of welded longitudinal seam of 90 per cent of that of the solid plate, in computing its wall thickness.

For service conditions of moderate severity, a pressure vessel may be constructed in accordance with the rules of paragraph U-69 (Class II). The following limitations, however, apply to this class, the exceeding of any of which makes mandatory the adherence to the rules of paragraph U-68 as summarized above.

a. The vessel must not be used for the storage of lethal gases or liquids (hydrocyanic acid, carbonyl chloride cyanogen, mustard gas and xylyl bromide)
b. The vessel must not be used for the storage of liquids under pressure at temperatures over 300 degrees Fahr.
c. The maximum working pressure cannot exceed 400 pounds per square inch
d. The maximum gas temperature cannot exceed 700 degrees Fahr.
e. The shell thickness cannot exceed 1½ inches.

Under this class of construction the welders must also have shown their ability to comply with the requirements of good welding by periodic tests. No X-ray of the welds are made, nor are test plates prepared. Vessels are stress relieved only when:

a. The plate thickness exceeds 1¼ inches
b. Both the wall thickness is greater than .58-inch and the shell diameter is less than 20 inches

(Continued on Page 124)

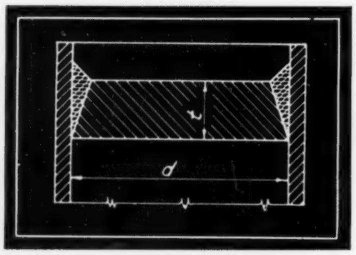
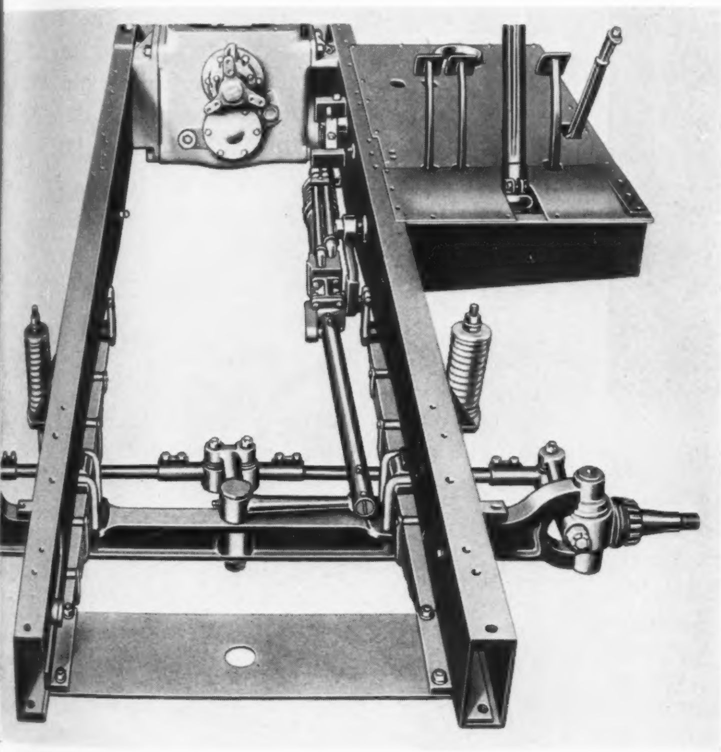


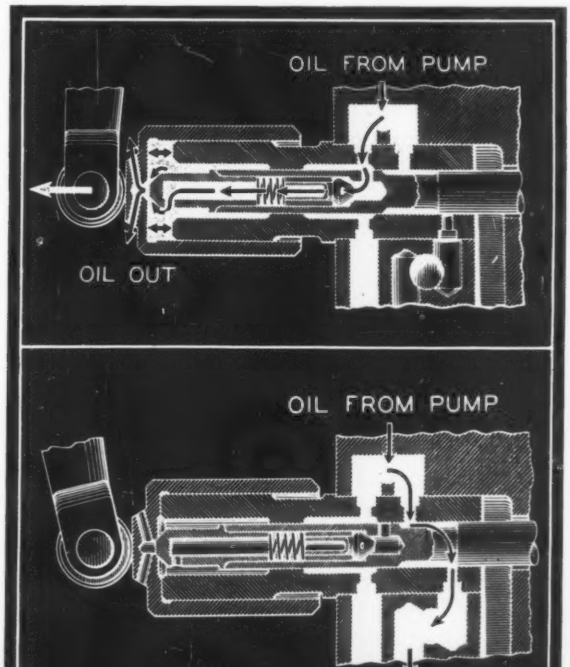
Fig. 7—For small diameters, a flat head welded from the outside is satisfactory

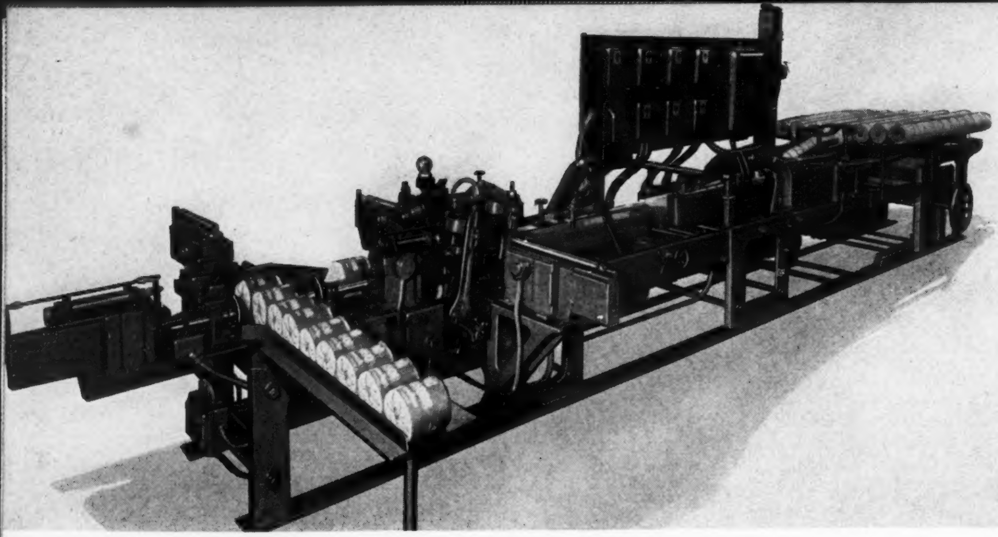
Scanning the Field FOR IDEAS



Finger-tip control through the use of a hydraulic booster reduces steering effort on the Caterpillar tractor assembly illustrated. Twin valves are so designed that they admit hydraulic pressure to a cylinder connected to the steering arm when the operator applies pressure to the steering wheel in either direction. The hydraulic pressure assists him as indicated by the drawings.

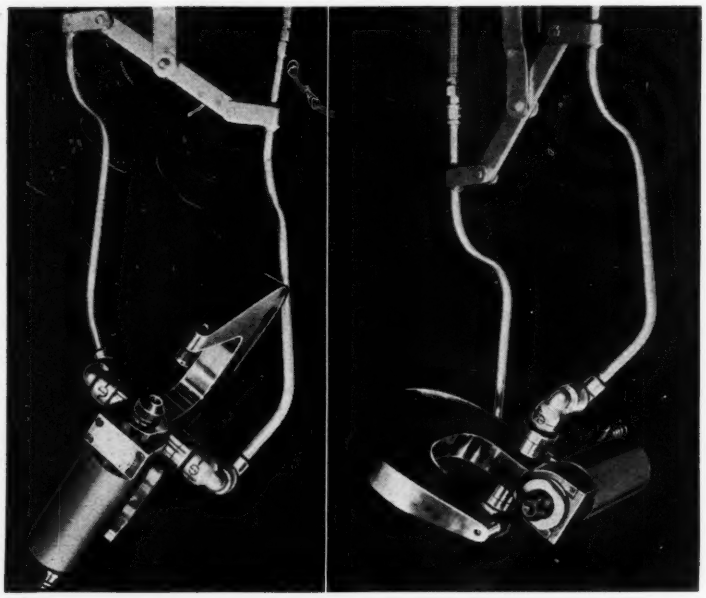
Attached to the lower end of the steering arm is a hydraulic ram controlled by the valves which are attached also to the steering arm. Pressure is admitted at one side of the ram for steering in one direction and at the opposite end for the other. When the operator discontinues turning the steering wheel, the valve immediately returns to neutral as shown in the drawings. Thus the operator handles the steering wheel in a normal manner but only has to apply sufficient pressure to open the valve. If for any reason there is no hydraulic pressure, the tractor can be steered manually.





Continuous, accurate operation of wrapping machine shown at left is effectively and easily controlled by the use of solenoids to obviate mechanical features which would slow down operation. Designed by Hudson Sharp Machine Co., the machine utilizes six Trombetta solenoids for various motions.

In operation, one solenoid feeds forward the roll of tissue to be wrapped. Two solenoids next in line operate cut-off knives to cut the wrapping paper between the tissue rolls after they have been twisted in twisting heads. As the wrapped roll moves into position, it touches a solenoid lever which releases two solenoid-operated plungers. These tuck the twisted ends in the core of the roll while the last solenoid, set at a right angle, places the wrapped package in delivery position and thus makes room for the next to enter.



Flexibility and ease of operation are provided by the riveter, above, through the combination of air hose with yoke having double-jointed universal connections. With this device, the operator is not handicapped by a conventional hose connection, enabling him to turn or twist the riveter into difficult positions with ease. Flexibility of movement provided by the combination yoke, designed by Hanna Engineering Works, further facilitates the handling of the unit.

Restricted air movement is provided within Frigidaire units by circulating the refrigerant through the walls in addition to through the evaporator in which ice cubes are made as shown at right. This reduced movement is obtained because no heat enters through the walls of the unit, since it is dissipated at the walls by the circulating refrigerant. Restriction of air movement, also is effected by sealed glass shelves which act as baffles, thus preventing absorption of moisture from the food.

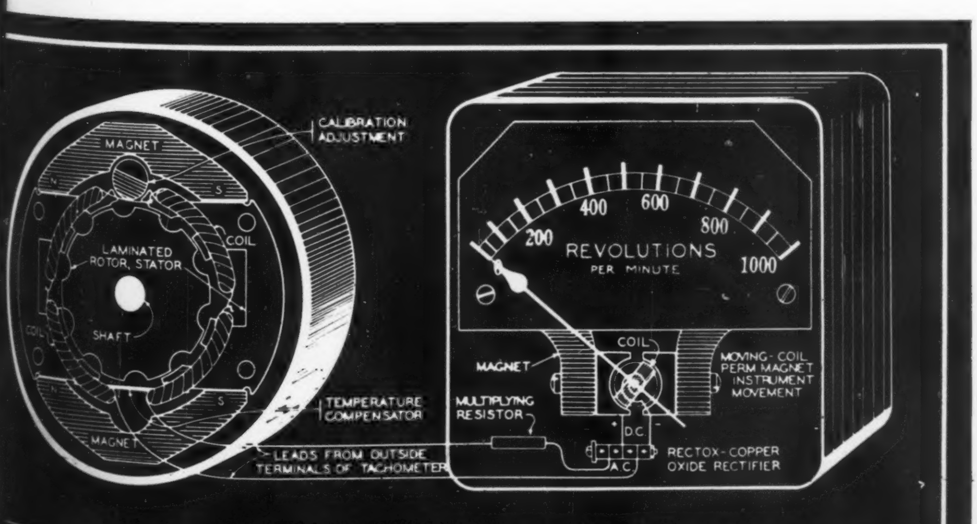
Photographic steel sheets simplify production of templates, nameplates, etc. Developed by Eastman Kodak Co., the sensitizing process consists of laminating a matte transfer film on the steel as shown in the photograph, right, taken at the Lockheed Aircraft Corp. plant. Engineering drawings may be printed either by contact or projection on the sheets. In this way, costly layouts are avoided as well as duplicate inspection. It is claimed that drawings may be projected and printed within a tolerance of .001-inch per foot. However, to maintain such accuracy it is necessary to select and utilize lenses with extreme care. To provide for correction, the film has a matte surface that will take pencil.

For contact prints, drawings are made on metal plates which have a coating that is fluorescent in the presence of X-rays. Either positive or mirror prints may be made by X-ray exposure through the back of the plate. Positive prints require an intermediate glass plate image. A mirror image, however, may be made into a "right" image by simply turning over the finished template.

Maintained calibration of tachometers and other measuring devices depend upon the permanence and simplicity of the operating mechanisms. To achieve this continued accuracy and reduce maintenance, Westinghouse engineers designed the tachometer shown below.

Utilizing an alternating-current generator, the tachometer has no brushes or commutator to cause errors due to changing brush contact or commutator wear. The generator consists of two permanent magnets, two laminated pole pieces, four coils, and a laminated rotor pivoted on ball bearings. Magnets are high-coercive magnet steel and receive two heat treatments, one to stabilize the crystalline structure and eliminate metallurgical aging and the other to age the material magnetically to reduce further changes. Mechanical hammers also shock the magnets to give them additional stability.

Parts are mounted in brass frame and the entire assembly is in an iron case to shield it magnetically. External disturbances, therefore, do not affect the calibration.



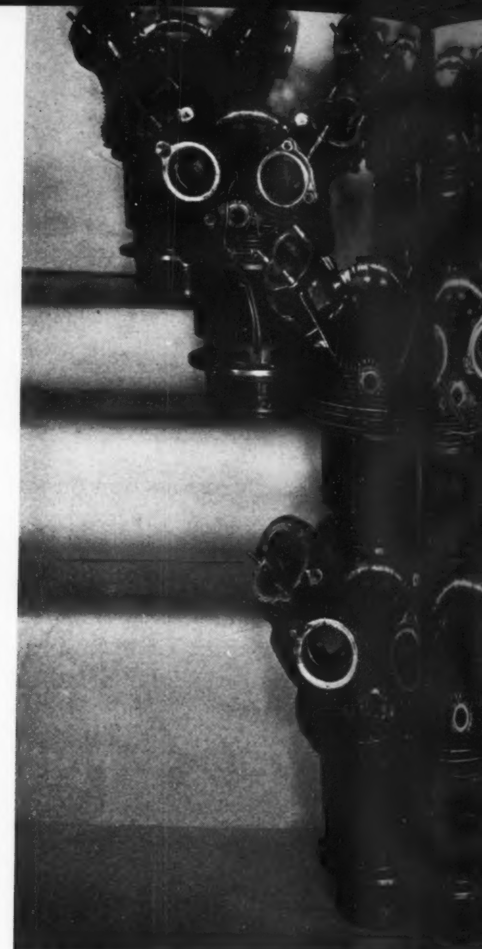


Fig. 1—Pratt and Whitney radial engine cylinders are rust and corrosion resistant in consequence of porous chromium plating application

Bearing Life Increased by New Finish

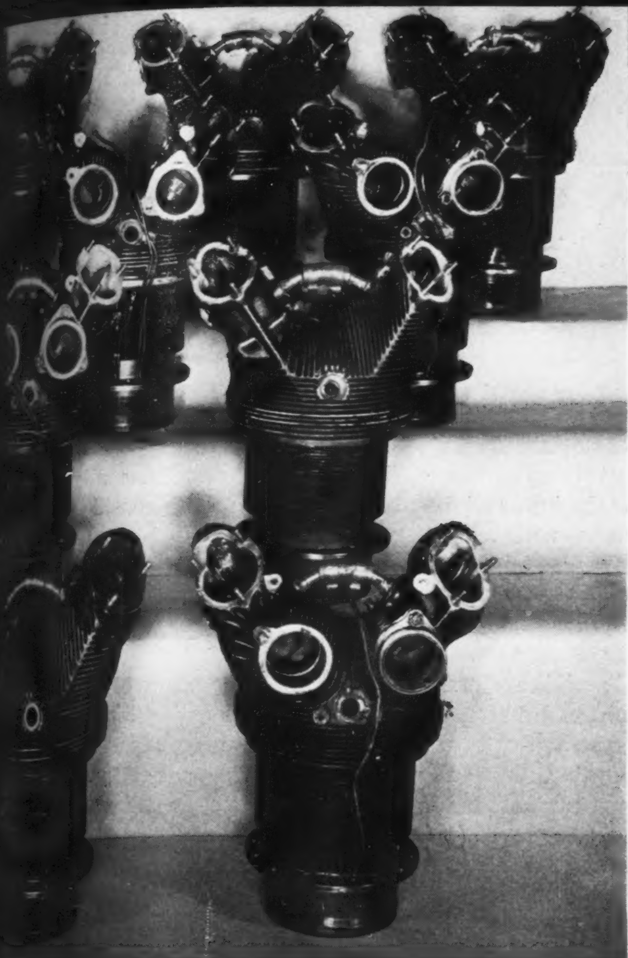
By H. Van der Horst
Van der Horst Corp. of America

BECAUSE of its hardness and resistance to abrasion, chromium has long afforded an intriguing possibility for the solution of difficult bearing problems. However, the disadvantages of the smooth, bright surface resulting from conventional plating procedures seemed to offer insurmountable difficulties. Oil will not "wet" this surface; its extreme smoothness departs from the generally recognized ideal bearing which possesses minor surface irregularities. The recent development of a method for depositing, electrolytically, a porous plating of chromium on bearing surfaces overcomes these objections and affords an extremely high resistance to wear, corrosion and abrasion.

Nowhere is the ever present bearing problem in machines more acute than between the piston rings and cylinder walls of internal combustion engines; and of these, the two-cycle diesel is the most formidable. It is in such engines as these, burning the cheapest of fuels and operating under the poorest lubrication conditions, that the advantages of the special porous chromium-surfaced cylinders have been strikingly evidenced.

Proved satisfactory in this service, the application of the process has already been extended to

Fig. 2 — Chrome - hardened cylinders in bus engine blocks wear only .002—inch in 100,000 miles. Ring wear is reduced to one-fifth the former amount



radial airplane engine cylinders, *Fig.* 1, bus engine cylinders, *Fig.* 2, and also gasoline and diesel marine engines, superheated steam engines, crankshaft bearings, crosshead pins, valve guides, machine tool ways, etc., with consistently good results.

What are the physical characteristics of this surface? It is full of microscopic pits and pores so that its density is about 30 per cent less than dense chromium. It has a hardness of about 1400 brinell. As mentioned previously, it possesses the characteristic roughness of a good bearing surface. Under conditions of even the poorest lubrication it has a remarkably low coefficient of friction with cast iron.

Chromium itself is brittle and has little cohesion. Superficially, this might lead to the conclusion that chrome-hardened bearings would be subject to failure by fatigue. However, because of the close bond or adhesion of the plating, the property of fatigue resistance is a function of the base metal upon which the chromium is deposited. Resistance to fatigue increases as the hardness of the backing metal increases, approaching the ultimate when hardened steel is used. Suddenly applied loads, such as those experienced in properly fitted crankshaft and wristpin bearings, are carried successfully. Impact loading, however, will, because of the poor cohesiveness of the material, result in early failure.

As part of a program of investigation of cylinder and piston ring materials, tests have been carried out on a watercooled cylinder barrel, chomium plated. The operating cycle consisted of 5 minutes idling and 10 minutes under load followed by 15 minutes cooling. The normal amount of lubricant was supplied.

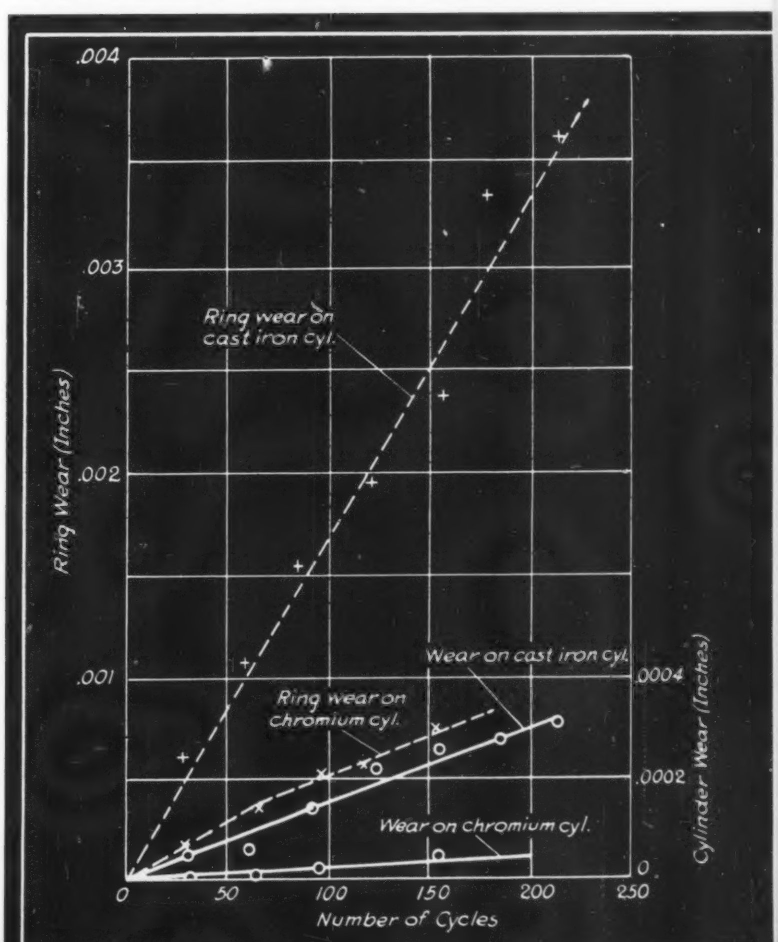
Normal rate of wear was first determined on a plain cast iron cylinder of 241 brinell hardness. The chrome-surfaced cylinder was then substituted for the cast iron using the same piston and rings and under the same conditions. Wear data is shown plotted in *Fig.* 3 for both the cylinder and the top piston ring.

The rate of wear of the plain cast iron cylinder as extrapolated from this curve is approximately .00022-inch per 1000 miles—a fairly normal rate. Wear on the chromium plated cylinder, as shown in the graph, is so low as to make a similar extrapolation impractical.

Severity of the tests was increased by withholding all oil supply during the preliminary 5-minute idling period of each 15-minute run so that the only lubrication was that left on the cylinder walls from the previous run. Results of this test are shown in TABLE I. These figures indicate extremely rapid wear in the plain cylinder, corresponding to .001-inch per 260 miles. As before, the wear of the chromium-plated cylinder was small and could not be measured accurately but was of the order of 1/25 that of the plain cylinder. In addition, there was a corresponding reduction in piston ring wear which the figures indicate to range between 10:1 and 30:1.

A further series of tests was conducted in which

Fig. 3—Results of comparative tests on standard cast iron cylinders and chrome-hardened cylinders indicate magnitude of wear reduction of both cylinders and rings



the rings were plated (chromium must not be applied to both pistons and rings or failure will be almost immediate). These tests involved continuous running in an aircooled engine at 1600 revolutions per minute. As before, the normal rate of wear was determined by using sand-cast rings having a brinell hardness of 290. TABLE II summarizes the results. These figures show a reduction of ring wear in the ratio of 3.7:1 and of cylinder wear of 2.8:1.

Results of these tests are the most convincing refutation of the thesis that the wearing qualities of the chromium surface stem from its hardness. If this were true, no diminution of wear on the unplated co-operating surface could reasonably be expected. Indeed, because of the extreme hardness of the chromium, an increase in this wear would not be irrational. Actually, the converse is true. Therefore the wear resistance of the chromium surface can largely be attributed to the peculiar surface properties of the material.

The question can be logically raised as to how a chrome-hardened surface compares with hardened cylinder liners. In 1937, the British Fuel Research Station conducted a series of tests in an engine using coal dust mixed with a little gasoline as a fuel. Using standard cast iron liners the wear was enormous. Also, the nature of the wear was different; instead of being all at the top, the bore was worn equally throughout the ring travel.

TABLE I

Wear with Plated Cylinders

Cylinder Condition	Cylinder Wear (in. per 1000 miles)	Ring Wear (in. per 1000 miles)
Plain cylinder	.0031	.0274
Plated cylinder	.00015	.0031
Plain cylinder	.0046	.085

Nitrided liners of about 1000 brinell hardness and porous chrome-hardened liners were tried in this engine with the following results: Chromium was 70 times better than cast iron and appreciably better than the nitrided liners. The wear on the chromium was in the order of .002-inch for 5400 hours or, expressed in mileage, for about 180,000 miles. It is hard to conceive of more severe abrasive conditions. Yet difference in hardness alone is insufficient to account for the wearing advantage.

As a general rule the wear on porous chromium will be about one-tenth that on cast iron. However, if service conditions are ideal this wear may be only one-fourth; with low grade lubricants, abrasive conditions and potential corrosion, wear on chromium may be as low as 1/20.

Difficulties currently experienced in the corrosion failure of leaded bronze bearings, especially in heavy duty automotive internal combustion engines, may be overcome by chromium treatment. Such a solution is particularly desirable in that the only proved alternative is tin-base babbitt. Because of the lower load capacity of such bearings designers are confronted, in their use, by the necessity of increasing engine weight or sacrificing torsional rigidity.

In designing for the application of porous chromium plating on any bearing surface there are several important considerations. First of these is the thickness of plating necessary. This decision is based solely on the type of service for which the bearing is intended. Normal rate of wear, conditions of abrasion and corrosion, desired service life, load characteristics, etc., are factors. Normally, the thickness of the plating based on these considerations will vary between .006 and .040-inch as measured on the diameter.

Sound Castings Essential

In the case of bearings or cylinder bores of cast iron, whether liners or engine blocks, variation in the structure of the iron, resulting in weak surface spots, will lead inevitably to the failure of the plating. All possible precautions should be taken to mitigate these defects. It is small consolation to rationalize after failure occurs that even without the plating it would have been a bad bearing anyway.

The question naturally arises as to what major design changes are necessary to utilize the plating. The answer is none at all. Subsequent experience may indicate that, in cast iron, unnecessarily close clearances may have been used so that satisfactory life would be obtained even after appreciable wear. It is therefore possible that, to some extent, such tolerances may be ultimately relaxed without sacrifice of service life. But in the initial design, such compensatory changes are inadvisable.

Cost is always a vital consideration. In this regard one large manufacturer of diesel engines for trucks, which formerly utilized dry sleeve liners, now uses the cheapest cast iron for their blocks. In so doing, the cost of the engine with chrome bores is actually less than formerly when liners were used.

In several respects the chromium plating of the rings only may afford a more expeditious utilization of the plating. The design revisions are simpler, cheaper and—important today—less time consuming. In this regard it is important to point out that the wear of rings occurs more rapidly than that of cylinder bores and thus, in a correspondingly shorter period, the chromium will be worn through. However, replacement of rings is comparatively inexpensive and, considering the reduced cylinder wear, this disadvantage may be more than outweighed.

TABLE II

Wear with Plated Rings

Ring Condition	Cylinder Wear (in. per 1000 miles)	Ring Wear (in. per 1000 miles)
Sand-cast rings ...	.00007	.00082
Plated rings	.000025	.00022

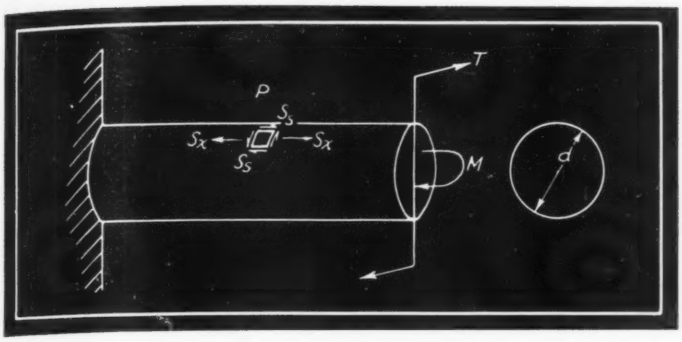


Fig. 1—Stresses produced at a point as the result of a bending moment combined with a twisting moment

Designing Shafting for Static or Fatigue Loads

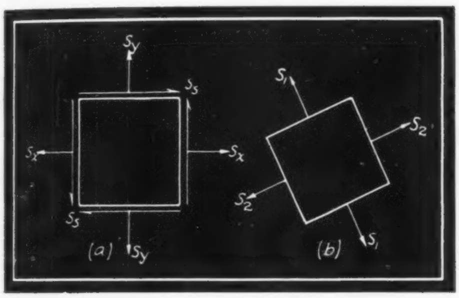


Fig. 2—Normal stresses and shearing stresses at (a) are shown replaced by principal stresses at (b)

By Joseph Marin
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RECENT research on the behavior of metals under various load conditions is particularly useful in the design of shafting for machine parts. The usual procedure has been to use the A. S. M. E. code for the design of transmission shafting. This code was published twelve years ago, however, and there has more recently come to light considerable information on the properties of materials subjected to combined stresses.

Theories are presented in the following which incorporate this new research on metals, and applications to specific problems in shaft design are then made. The differences between the old code and the new data are shown to be considerable in some cases. For the convenience of the designer the results are summarized in the form of design charts by means of which the diameter of the shaft can be selected for particular load combinations. For the case of fatigue loads a new theory is proposed based on test results.

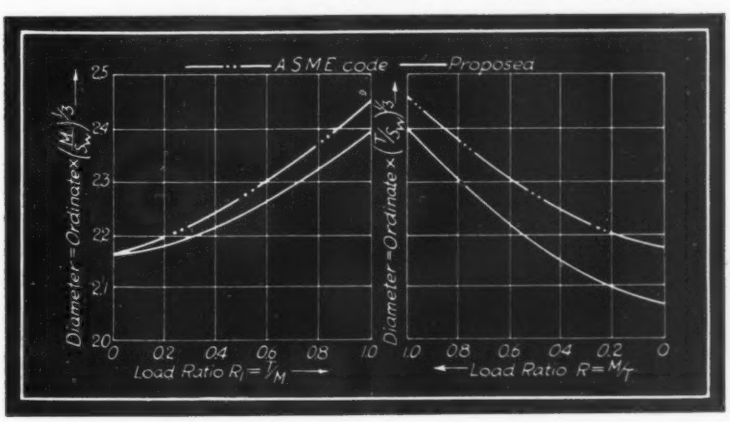
In both transmission shafting or shafts forming a part of a machine, a combination of loads is usually present as, for example, a bending moment M combined with a twisting moment T, *Fig.* 1. The stresses produced by these moments on the outer fiber of the shaft at a point P are

$$S_b = \frac{32M}{\pi d^3} \quad , \quad S_s = \frac{16T}{\pi d^3} \quad \ldots\ldots\ldots\ldots\ldots (1)$$

where d = the diameter of the shaft.

In determining the shaft diameter which will resist the bending and twisting moments it is necessary to consider the mutual effects of the stresses in producing failure in Equation 1. In order to do this, experiments have been made to study the be-

Fig. 3—Static bending and torsion curves for comparing A.S.M.E. code with the proposed theory



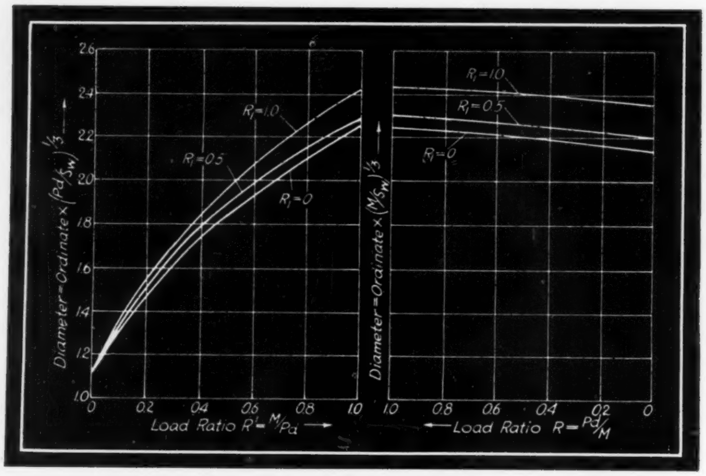


Fig. 4a—Static bending, torsion and axial load curves for A.S.M.E. code and the proposal. This chart provides a convenient means for selecting shaft diameter

havior of materials subjected to combined stresses. Theories have also been advanced to represent these test results and to express the behavior of materials subjected to a general system of combined stresses.

Both static and fatigue loadings will be considered in the following. For both conditions the A.S.M.E. code and the theory developed in the article will be compared. In the case of fatigue the new theory based on tests is given for the first time.

In discussing the theories of failure for materials subjected to combined stresses it is convenient to designate the stress components by their principal stresses. For the case of two-dimensional stresses or stresses acting in one plane the stress components at a point in a stressed member are represented by normal stresses S_x and S_y and a shearing stress S_s as shown in *Fig. 2a*. It is convenient to replace these stresses by the principal stresses S_1 and S_2 as shown in *Fig. 2b*. In this way all systems of stresses are reduced to a common system in which the stresses act normal to planes which are free from shear stresses. The principal stresses also represent the maximum and minimum values of the normal stresses[1]. It can be shown that the principal stresses S_1 and S_2 in terms of stress components, *Fig. 2*, are

$$S_1 = \left(\frac{S_x + S_y}{2}\right) + \sqrt{\left(\frac{S_x - S_y^2}{2}\right)^2 + S_s^2} \quad \ldots\ldots (2a)$$

$$S_2 = \left(\frac{S_x + S_y}{2}\right) - \sqrt{\left(\frac{S_x - S_y}{2}\right)^2 + S_s^2} \quad \ldots\ldots (2b)$$

Yielding or failure of the element in *Fig. 2b* will result for certain limiting values of the principal stresses. That is, for a particular value of the principal stress S_2 the value of S_1 producing failure will have some definite value. Confining the discussion to ductile materials, failure will be defined when yielding occurs. For simple tension this is defined by the lower yield point or the "proof" stress and will be designated by S_{yp}. Returning to the case of combined stresses as represented in *Fig. 2b*, the value of S_1 at which yielding occurs will be some quantity times S_{yp}—the relation between S_1 and S_{yp} depending upon the value of the stress S_2. The values of the principal stress S_1 at yield in terms of the stresses S_2 and S_{yp} have been determined experimentally for various materials. A number of theories have been proposed to explain these test results[2]. Only a brief statement of two of the theories will be made here.

The *Shear Theory* of failure is used considerably by machine designers for ductile materials and appears in several codes such as the Code for the War, Navy and Commerce Committee on Aircraft Requirements, the Westinghouse Co. code and the A.S.M.E. code for design of transmission shafting. According to the shear theory of failure, a material subjected to combined stresses fails or yields when the shear stress becomes equal to the shear stress at yield in simple tension. It can be shown[2] that this condition is defined by the equations

$$(S_1 - S_2) = \pm S_{yp} \quad \ldots\ldots (3a)$$

or

$$S_1 = \pm S_{yp}, \quad S_2 = \pm S_{yp} \quad \ldots\ldots (3b)$$

Equations 3*a* apply when the principal stresses are of opposite signs and Equations 3*b* for stresses of the same sign.

For many years test data have not been conclusive as to which theory of failure is best for ductile metals. Many tests indicate that the shear theory expressed by Equations 3 is in good agreement with experiments. In recent years, however, the distortion energy theory of failure has been verified for ductile metals[2].

The *distortion energy theory* was developed sev-

[1] For a discussion of principal stresses see, for example, F. B. Seeley, *Resistance of Materials*, John Wiley and Son.

[2] For a discussion of test results and theories of failure see A. Nadai, "Theories of Strength", *Trans.* A.S.M.E., 1933; J. Marin, "Failure Theories of Materials Subjected to Combined Stresses", *Trans.* A.S.C.E. Vol. 101, 1936; and J. Marin and R. L. Stanley, "Failure of Aluminum, Subjected to Combined Stresses", *Journal American Welding Soc.*, Feb. 1940.

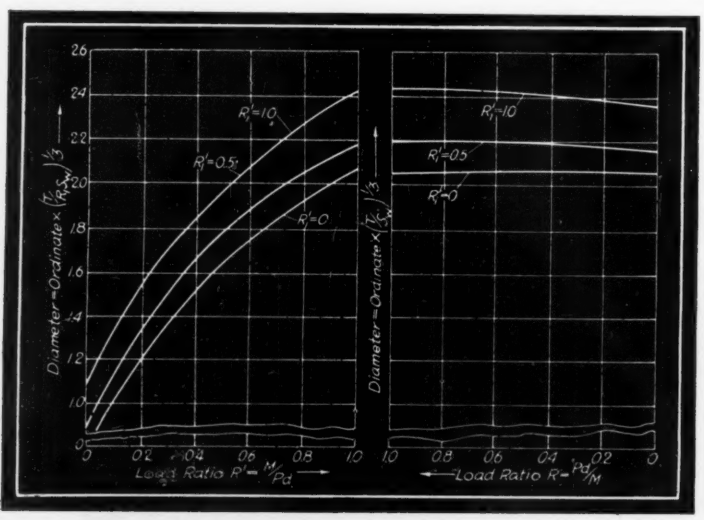


Fig. 4b—Comparative static bending, torsion and axial load curves plotted similarly to those in Fig. 4a

eral years ago by Von Mises[3] and Hencky[4]. It assumes that failure begins in the case of combined stresses when the energy of distortion or shear is equal to the energy of distortion in the case of simple tension. This theoretical basis[2] leads to the relation between the principal stresses at failure:

$$S_1^2 - S_1S_2 + S_2^2 = S_{yp}^2 \qquad (4)$$

The theory has been accepted by many investigators as representing failure in ductile metals subjected to combined stresses. It is of interest to show the differences resulting in design using this relationship in place of the shear theory. For this purpose two examples are considered in the next section.

To use Equations 3 and 4 for purposes of design it is only necessary to replace the yield point stress in simple tension, S_{yp}, by the working stress S_w. It is also convenient to express the above theories in terms of the stress components in place of the principal stresses by substituting S_1 and S_2 from Equations 2 in Equations 3 and 4. The allowable relationship between the stress components by each theory then becomes by

1. The Shear Theory:

$$S_w = \pm 2\sqrt{\left(\frac{S_x - S_y}{2}\right)^2 + S_s^2} \qquad (5a)$$

$$S_w = \left(\frac{S_x + S_y}{2}\right) \pm \sqrt{\left(\frac{S_x + S_y}{2}\right)^2 + S_s^2} \qquad (5b)$$

Equation 5*a* applies for stresses of the opposite signs and Equation 5*b* for stresses of the same sign.

2. The Energy Theory:

$$S_w^2 = S_x^2 - S_xS_y + S_y^2 + 3S_s^2 \qquad (6)$$

Significance of the above equations can be seen by considering a specific case in which the values of the stress components S_s and S_y are fixed in magnitude. Then for a material with an allowable stress in tension of S_w, the allowable value of the stress S_x is given by the value of S_x determined in Equation 6, or by Equation 5 if the shear theory is considered.

The two theories stated above represent in one case the theory usually used and in the other case the theory agreeing best with test results. With these as a basis, applications can be made to designs.

Example 1—Shaft Subjected to Static Bending and Torsion: A circular shaft subjected to torsion and bending is represented in *Fig.* 1. The critical element by inspection is, in this case, on the top or bottom fibers where the bending stress and torsion stress are both maximum and of values expressed by Equations 1. The value of the diameter required by the shear and energy theories is obtained by substituting the values of the stress components from Equations 1 in Equations 5 and 6. Doing this, the values for the required diameter by the shear and energy theories are respectively

$$d_1 = (2.17)\left(\frac{T}{S_w}\sqrt{R^2+1}\right)^{1/3}$$
$$d_2 = (2.17)\left(\frac{T}{S_w}\sqrt{R^2+.75}\right)^{1/3} \qquad (7)$$

where R = the load ratio M/T.

Variation in diameter based on the code and the proposal, for different values of the load ratios is shown in *Fig.* 3. In order to cover all possible combinations of bending and twisting moments, R in Equations 7 can be replaced by $1/R'$ where $R' = T/M$. This substitution changes Equations 7 to

$$d_1 = (2.17)\left(\frac{M}{S_w}\sqrt{1+(R')^2}\right)^{1/3}$$
$$d_2 = (2.17)\left(\frac{M}{S_w}\sqrt{1+.75(R')^2}\right)^{1/3} \qquad (7a)$$

By considering values of both R and R' from 0 to 1, all possible ratios of the bending to the twisting moment are included. The ordinates in *Fig.* 3 are proportional to the diameter of shaft required. That is, for a particular known load ratio the ordinate to the curve corresponding to the known value of R or R', multiplied by $(M/S_w)^{1/3}$ or $(T/S_w)^{1/3}$ is the shaft diameter. From *Fig.* 3 a comparison can be made. The maximum difference between the two theories is for pure torsion where $R = M/T =$

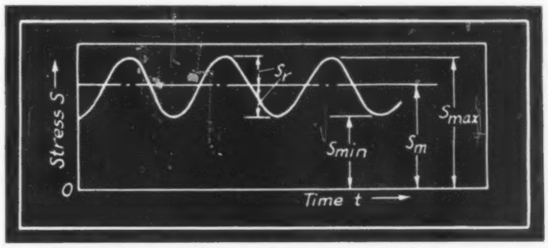
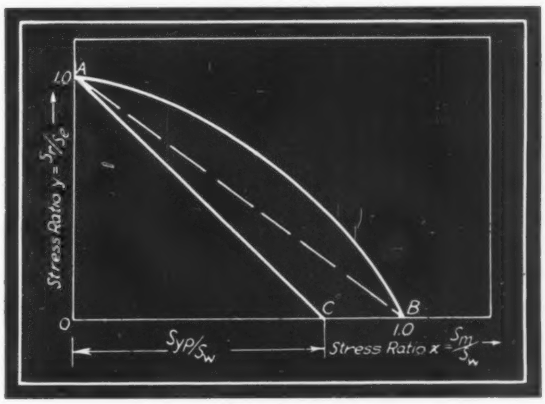


Fig. 5—General case of axial fluctuating stress

Fig. 6—Plotted results from tests in axial stress fatigue machine



[3] R. Von Mises, Gottinger Nachrichten, *Math. Phys.* Kl, 1913, p. 582.

[4] M. Hencky, "Zeitschr. F. Angew." *Math. und Mech.*, Vol. 5, 1925, p. 115.

0. The difference for this particular problem, however, is not appreciable.

EXAMPLE 2—SHAFT SUBJECTED TO STATIC AXIAL LOAD COMBINED WITH TORSION AND BENDING MOMENTS: Sometimes, in addition to the bending and twisting moments shown in *Fig.* 1, an axial load P is present. In some constructions this occurs when the shaft is vertically placed and P then represents a compressive load. In this case the critical element is at the lower fiber in *Fig.* 1 where the stress components are as given by Equation 1 except that the normal stress is modified due to the load P. The stresses are then

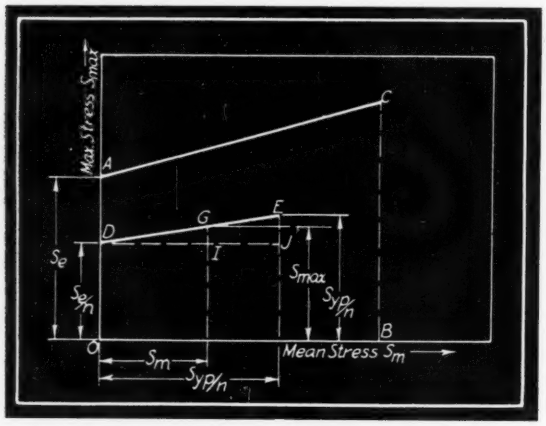


Fig. 7—Variations in maximum stress showing mean stress plotted against maximum stress

$$S_x = -\left(\frac{32M}{\pi d^3} + \frac{4P}{\pi d^2}\right) \quad , \quad S_s = \frac{16T}{\pi d^3} \ldots (8)$$

As in the previous example, the diameter of shaft required by the energy theory is determined by placing the stresses from Equation 8 in Equation 6. Doing this, the diameter required is the value of d in the equation

$$d^3 = (1.27)\left(\frac{M}{S_w}\right)\sqrt{(8+R)^2 + 48R_1^2} \ldots\ldots\ldots (9)$$

where $R = Pd/M$ and $R_1 = T/M$.

All possible values of the load ratios R and R_1 can be considered by replacing R by $1/R$ where $R' = M/Pd$ and R_1 by $1/R_1'$ where $R_1' = M/T$. By varying these load ratios from 0 to 1 the various possible combinations of the three loads M, T and P are included. The ordinates to the curves in *Fig.* 4*a* and 4*b* are proportional to the diameter of shafting required for each particular value of the load ratios. This design chart affords a convenient means of selecting the shaft diameter. Since the load ratios R and R' involve the quantity Pd it is necessary to assume a value of d and use a "trial and error" procedure to obtain the correct diameter. This does not necessarily lead to serious difficulties.

In the above two examples shear stresses due to transverse shear were not considered and the bending moments were assumed as pure bending moments. For most cases the stresses produced by the transverse shear are small compared to the other stresses and it can be shown that their influence on the shaft diameter is negligible. This is not true, however, in some cases as, for example, when the length of shaft is relatively small compared to the shaft diameter. This type of loading, as well as others, can be treated as in Examples 1 and 2. The analysis can also be made for certain shafts of noncircular cross section.

Designing for Fatigue Loads

Shafts used for transmitting power are subject to stresses which do not remain constant with time but which fluctuate in magnitude as the shaft rotates. The material under these conditions is subjected to a fatigue stress and in designing such shafts it is necessary to consider the fatigue strength of the material.

The A.S.M.E. code for the design of transmission shafting which was previously referred to considers the decrease in strength of the material under a fatigue stress condition by assuming, for example, that if the shaft in *Fig.* 1 is rotating, the bending moment is equivalent to a static bending moment equal to a constant k times the actual moment. For gradually applied and steady loads on a rotating shaft the factor k is given as 1.5. If the loads are applied as shock loads the code specifies a factor k up to 3.0. Since the stresses are directly proportional to the loads, the A.S.M.E. code can be defined, in terms of the stresses, by multiplying each stress component in Equation 5 for the shear theory by a factor k. The working stress values then become

$$S_w = \pm 2\sqrt{\left(\frac{k_x S_x - k_y S_y}{2}\right)^2 + (k_s S_s)^2} \ldots\ldots (10a)$$

$$S_w = \left(\frac{k_x S_x + k_y S_y}{2}\right) \pm \sqrt{\left(\frac{k_x S_x - k_y S_y}{2}\right)^2 + (k_s S_s)^2} \ldots\ldots (10b)$$

As in Equations 5, Equation 10*a* applies for stresses of opposite sign and Equations 10*b* for those of the same sign.

The above formula is the one that is usually used[5]. In this type of loading, as in static loading, however, research in recent years on the mechanical properties of materials in fatigue shows that considerable error in design results when the A.S.M.E. code is used. For this reason the writer discussed a theory[6] for failure of metals subject to combined fatigue stresses. This theory was found to agree

[5] The Westinghouse Elec. & Mfg. Co. uses a shear theory which considers the fatigue strength as an influencing factor. This is not the one discussed in the theory proposed. It is presented by C. R. Soderberg in "Working Stresses", A.S.M.E. *Trans.* Vol. 57, 1935, p. 106.

[6] J. Marin, "Working Stresses for Members Subjected to Fluctuating Loads", *Trans.* A.S.M.E., 1937, Vol. 59, p. 55.

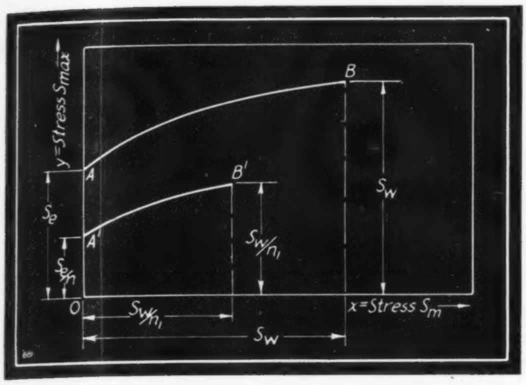


Fig. 8—Relationship between the maximum stress value and the mean stress value. Working stress curve is assumed same shape as failure curve AB

well with the test results available. A brief summary of this theory as well as a modified one not previously presented will now be outlined. For this purpose it is necessary first to consider the case of axial fluctuating stress.

For the general case of axial fluctuating stress the magnitude of the stress varies from a minimum value S_{min} to a maximum value S_{max} so that the mean value is S_m as shown in *Fig.* 5. This fluctuating stress is equivalent to a static stress S_m upon which is superimposed a variable stress S_r. That is shown by

$$S_m = \left(\frac{S_{max}+S_{min}}{2}\right), \qquad S_r = \left(\frac{S_{max}-S_{min}}{2}\right) \quad \text{..(11)}$$

Stress S_{max} represents the fatigue strength if this stress can be theoretically maintained for an infinite number of cycles of fluctuations. For a complete interpretation of fatigue strength test results it is necessary to consider all possible combinations of the maximum and minimum stresses. In other words it is necessary to consider the effect on the fatigue strength—defined by S_{max}—of the stress S_m or the variable stress S_r. This information has in some cases been obtained from an axial stress fatigue machine. It has been customary to plot the results of the tests on the coordinate axes shown in *Fig.* 6 where x is the stress ratio S_m/S_u (S_u being the ultimate stress in static tension) and y is the stress ratio S_r/S_e (S_e representing the fatigue strength for complete reversal of the stresses). Then point B represents static tension and point A complete reversal of stress—a stress such that the mean stress as shown in *Fig.* 5 is zero. Other combinations of the mean and variable stresses can be represented by coordinates between the points A and B.

Empirical relations have been proposed in the past to define failure for intermediate points between A and B. Only two will be considered here. One method which has often been advocated[7] is to assume a straight line between a point C representing the yield point and the point A. Test results have been presented which show that this line is on the safe side of the test results. The equation of the line AC is

$$S_r = S_e - p\,S_m \quad \text{..........(12)}$$

where $p = S_e/S_{yp}$

For the purpose of defining working stresses it is convenient to write Equation 12 in terms of S_{max} and S_m in place of S_r and S_m by substituting S_{max}—S_m for S_r in Equation 12. Doing this

$$S_{max} = (1-p)S_m + S_e \quad \text{..........(13)}$$

The variation in the maximum stress, S_{max}, is shown in *Fig.* 7 which is a plot of Equation 13. Points A and C represent complete reversal and static loadings, respectively. The allowable stress values corresponding to these points A and C are points D and E, respectively. The working stress formula showing the line DE is obtained by noting that $GI/EJ = DI/DJ$ or

$$S_{max} = S_m(1-p) + \frac{S_e}{n} \quad \text{..........(14)}$$

where n = the factor of safety.

If S_w is the working stress in simple tension then

$$S_w = \frac{S_{yp}}{n} = \frac{S_e}{pn} \quad \text{..........(15)}$$

Placing the value of S_e/n from Equation 15 in Equation 14 and solving for the working stress

$$S_w = \frac{1}{p}\left[S_{max} - S_m(1-p)\right] \quad \text{..........(16)}$$

In this way the working stress, S_w, is defined in terms of the maximum and mean stresses.

Another relationship between the maximum and

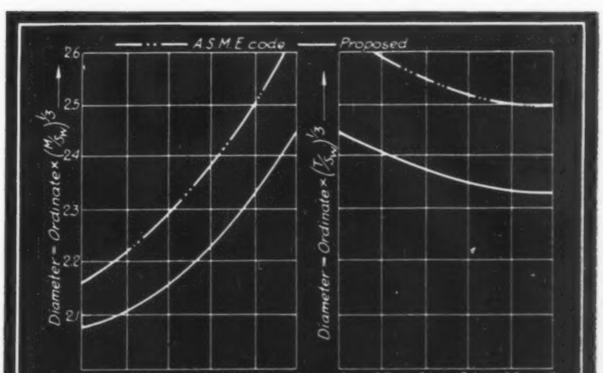


Fig. 9—Load ratios for rotating shaft subjected to both bending and torsion

[7] See for example J. M. Lessells, "Symposium on Significance of the Tension Test of Metals in Relation to Design", A.S.T.M. *Proc.*, Vol. 40, 1940, p. 501.

mean stresses which has experimental support is based on a parabolic relationship indicated in *Fig.* 6 by the curved line *AB*. The Equation of *AB* is

$$\left(\frac{S_r}{S_e}\right) = 1 - \left(\frac{S_m}{S_u}\right)^2 \quad \ldots\ldots (17)$$

This equation can be expressed in terms of the maximum stress by substituting $S_{max} - S_m$ for S_r. The value of the maximum stress S_{max} then becomes

$$S_{max} = S_e + S_m - S_e\left(\frac{S_m}{S_u}\right)^2 \quad \ldots\ldots (18)$$

This relationship, expressed by Equation 18, between the stresses S_{max} and S_m is plotted as the line *AB* in *Fig.* 8. Assuming the working stress curve

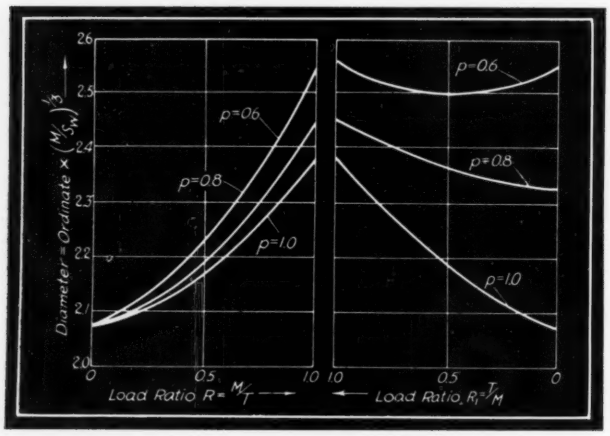


Fig. 10—Effect of varying the material constant on load ratios for shafts under bending and torsion

A'B' is similar in shape to the failure stress curve *AB*, a working stress equation for *A'B'* can be obtained to replace Equation 18. In doing this the same factor of safety will be used for the yield point as for the endurance limit S_e. Then

$$\frac{S_u}{n_1} = \frac{S_{yp}}{n} \text{ and } n_1 = \left(\frac{S_u}{S_{yp}}\right) n \quad \ldots\ldots (19)$$

Above analysis shows that the working stress, in terms of the maximum and mean stresses, is

$$S_w = \left(\frac{1}{2p}\right)\Big[(S_{max} - S_m) + \sqrt{(S_{max})^2 - 2S_{max}S_m + S_m^2(1+4p^2)}\Big] \quad \ldots (20)$$

The working stress value has been expressed in both Equations 16 and 20, based on the failure stress relations *AC* and *AB*, respectively, *Fig.* 6. With these allowable stress equations established for axial stress, the more general case of combined stress can be discussed.

Theory of failure for combined stresses which is based on Equation 16 was developed by the writer[6] and shown to agree with the test results available on combined fatigue stresses. This theory is designated as the distortion energy theory and is based on the assumption that failure is a function of the distortion or shear energy. The maximum and mean values of the stress components are also assumed as occurring at the same instant of time. In developing this theory it is considered that failure occurs when combined stresses are present if the distortion energy corresponding to the maxi-

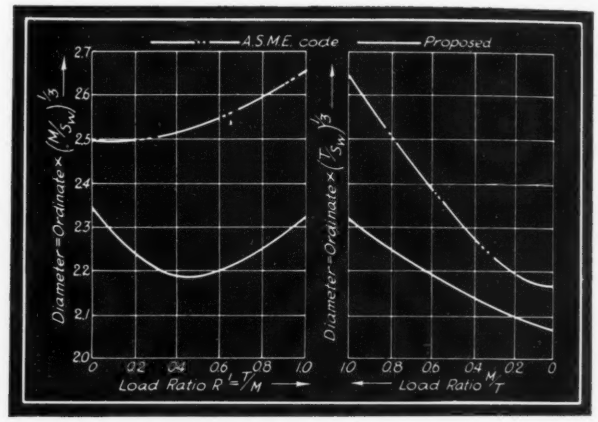


Fig. 11—Load ratio curves for failure condition defined by line AB in Fig. 6

mum values of the stress components equals the distortion energy at failure corresponding to the maximum axial stress. In these cases of combined and axial stresses equal energies corresponding to the mean stresses are considered.

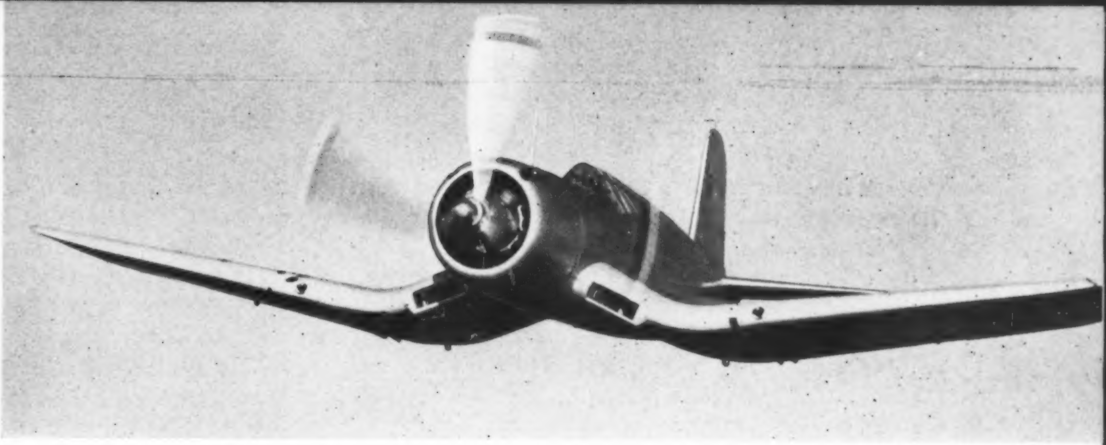
For a two-dimensional or plane state of combined stress, as shown in *Fig.* 2, let S_x', S_y' and S_s' be the maximum stress components and S_x'', S_y'' and S_s'' be the mean stress components. Then it can be shown, on the basis of the foregoing conditions of failure, that the working stress is

$$S_w = \frac{1}{p}\Big[\sqrt{(S_x')^2 - S_x'S_y' + (S_y')^2 + 3(S_s')^2} - (1-p)\sqrt{(S_x'')^2 - S_x''S_y'' + (S_y'')^2 + 3(S_s'')^2}\Big] \quad \ldots (21)$$

In a manner similar to that explained, a relation corresponding to Equation 21 can be derived based on the failure stress relation *AB* in place of *AC*, *Fig.* 6. This theory gives for the working stress

$$S_w = \left(\frac{1}{2p}\right)\Big\{\sqrt{(S_x')^2 - S_x'S_y' + (S_y')^2 + 3(S_s')^2} - \sqrt{(S_x'')^2 - S_x''S_y'' + (S_y'')^2 + 3(S_s'')^2} + \sqrt{(S_x')^2 - S_x'S_y' + (S_y')^2 + 3(S_s')^2 - 2[(S_x')^2 - S_x'S_y' + (S_y')^2 + 3(S_s')^2]^{1/2} \times [(S_x'')^2 - S_x''S_b'' + (S_y'')^2 + 3(S_s'')^2]^{1/2} + [S_x'')^2 - S_x''S_y'' + (S_y'')^2 + 3(S_s'')^2][1+4p^2]}\Big\} \quad . (22)$$

(*Continued on Page* 132)



Reputed to be the fastest in the United States, the Vought Fightee and other military planes are requiring the major portion of the aluminum output

Restrictions Challenge Ingenuity of Machine Designers

By Kenneth D. Moslander

SITUATIONS arising out of the defense effort present many and complex problems to the designer of machines. Of immediate concern is the scarcity of materials and the consequent necessity of finding substitutes or alternatives. Even when this has been done, the substitution in nearly all cases makes necessary some imperative design changes which may be of major importance.

These are everyday problems. So accustomed has the engineer become to such constant challenges to his ability and ingenuity that their solution seems almost routine. Once, each one would have been a nightmare.

Shortage of engineering manpower is acute, particularly men with the specialized training necessary to attack a problem and solve it without the prerequisite of a long training program. Plenty of men with sound academic backgrounds are available. However, when one of these is confronted with the task of designing a valve mechanism, for example, with its many ramifications of load, inertia, heat dissipation, corrosion resistance, pressure drop, etc., he is lost for lack of intimate, specialized knowledge.

A few companies, mostly larger ones, have for several years had elaborate training programs for engineering as well as shop personnel. These companies are a fortunate few. For the others, the eleventh hour has come. Upon the men who could conduct such training courses, the unprecedented load of current work has fallen. These men, of necessity, carry the burden of the engineering aspects of the defense effort.

Problems of engineering organization and expansion further test the resources of engineers. Less than 200 companies in the machine tool industry, in normal times, do 90 per cent of the average annual business volume of 100 million dollars. In producing the equivalent of this amount in machine tools less than 50,000 men are required. This year, with little more than twice as many men, production is expected to approach a 700 million dollar volume. The engineering job involved in this expansion is of first order magnitude; and the work is just beginning.

Expanded without Time Loss

When Pratt and Whitney completed the addition of 400,000 square feet of floor space in four new buildings for engine assembly the changeover from the old to the new buildings was accomplished over a weekend, without the loss of an hour. Workmen in the assembly plant left their jobs in the old plant Friday night and returned to their jobs in the new buildings Monday morning. Two days later a record high in daily output of engines was established. That this was a job requiring the ultimate in engineering planning and organization cannot be disputed.

Not all problems lend themselves to such effective solution. Difficulties ordinarily beget difficulties. Because of priorities on zinc and aluminum,

for instance, automotive engineers are faced with the need for new material specifications for die-cast carburetors. Cast iron affords a likely possibility. To make carburetors of cast iron requires a large increase in the number of machining operations. Machine tools for these operations cannot be obtained. Thus, a practical solution from the standpoint of materials priorities becomes impractical from the standpoint of equipment.

Thermosetting plastic resins might work. It might even be possible, with minor changes, to use the same dies as those in which the zinc alloy was cast. Conventionally, however, the carburetor supports, in whole or part, the not inconsiderable weight of the air filter. This, combined with normal engine vibration and road shocks, raises some doubts as to the material's physical properties. Reduction of inertia, removal of loads and the possibility of built-in steel reinforcing are all distinctly separate design problems upon which the final solution is contingent.

Continued recounting of problems and hardships would be as interminable as it would be valueless. What, specifically, can be done to alleviate material scarcity or to circumvent it where it cannot be alleviated? To this end "priority" materials may be considered separately.

Materials Considered Separately

ALUMINUM—The outlook as regards this metal is bright. The immediate pinch, felt early in the defense effort by users of other metals, was not so acute in the case of aluminum. This was largely due to the fact that the Aluminum Co. during the depression years, maintained its production in excess of sales in order to avoid excessive unemployment. The stock thus built up tided over the demand until recently. The breathing spell permitted expansion of facilities which by 1942 are expected to be adequate for both defense and civilian needs.

Just now production is beginning at the rate of 60 million pounds annually at the new plant of the Reynolds Metal Co. In 1939 the Aluminum Co. produced 325 million pounds. Present capacity is 450 million. With plant expansions already under way the total will reach 700 million by July, 1942.

Whereas slightly over half of the bauxite now used is imported from Dutch Guiana, it is estimated that even if this were interrupted the deposits in Arkansas would take care of requirements for at least 8 years.

Metallic substitutes for aluminum for applications wherein it is utilized for its lightness are, with the exception of magnesium, nonexistent. Further, the magnesium situation is more acute than that of aluminum. For castings such as automobile engine pistons it may be possible to use low grade aluminum. This material contains no virgin aluminum but is melted entirely from scrap. Containing varying percentages of copper, zinc, nickel, tin, manganese and bismuth, difficulties attendent on its drawing and forming are accountable for its present B-4 priority rating which permits 90 per cent of civilian orders to be filled.

The alternative as far as pistons are concerned is cast iron. Because of the increase of reciprocating masses with a consequent increase of bearing and connecting rod loads, requiring lower speeds and lower compression ratios as well as differences in thermal expansion rates, engineers will accept this alternative reluctantly.

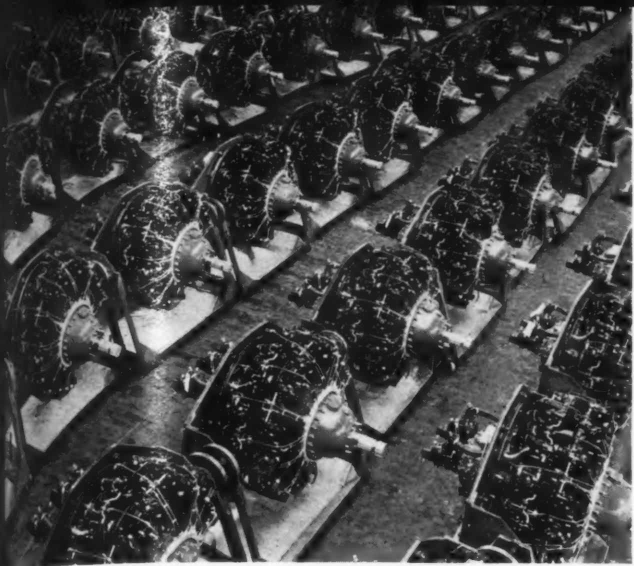
In many applications where weight is not so vital, steel presents a possibility. It should be remembered that the strength-weight ratio of aluminum to steel appreciably exceeds the ratio of densities.

Plastics Problems Solvable

Various plastics afford a second alternative. Here strength-weight ratios and impact strength approach the values of aluminum itself. Problems of dimensional instability particularly at elevated temperatures, brittleness at low temperatures and the necessity for providing adequate sections to carry the imposed loads can, in many cases, be "designed around." Steel reinforcing, sprayed and plated metal coatings and attention to shapes and contours in design will go far toward overcoming these objections. It is true that the location of holes particularly for screw fastenings requires special consideration. Here again, redesign and provision for inserts may afford a satisfactory solution.

ANTIMONY—The vast bulk of the antimony used in this country comes from Mexico. No immediate shortage of this material is anticipated. If one should develop ultimately, the die casting situation as regards zinc or aluminum should be cleared up by that time and no hardship need be expected.

Manganese steel, hard-case armor plate is used extensively in light tanks now in mass production



Emphasizing the demand for high-alloy, high-strength materials are these Pratt and Whitney engines, ready for shipment

As far as bearings are concerned, leaded bronze, sintered bronze and even substantially antimony-free, tin-base or lead-base babbitt may provide more than adequate alternatives.

CHROMIUM—By far the largest part of the chromium consumed is for alloy steels. While mandatory priorities have not been invoked, inasmuch as it comes from southern Europe and Africa, such priorities are a contingency to be anticipated. Used largely in the stainless steels for corrosion resistance and in other steels for deep hardening properties, its curtailment—particularly since nickel is not generally available—will be a hardship. For hardening and wear resistance properties the obtainability of molybdenum, vanadium and manganese will provide adequate substitutes. For corrosion resistance alone, recourse may be had to plating of either cadmium, chromium or zinc.

Little Chromium Used for Trim

The amount of chromium used for ornamental trim is so small as to be negligible; its deposited thickness is but little more than 10 millionths of an inch and sometimes less. Chrome trim is more likely to be eliminated because of nickel or copper shortages since either of these two provide the base plating over steel.

Steadily decreasing prices of indium offer an intriguing alternative. Indium is tarnish-free and corrosion resistant. True, it must be baked after application to obtain the necessary solid solution. But when this is done a bright lustrous surface is obtained which is not subject to peeling. If only nominal wear is experienced the finish will be practically everlasting.

COPPER—Our own production plus that imported from South America is more than adequate for defense needs. In addition, there are extensive low grade deposits here which may be worked profitably for a somewhat higher price per pound.

For electrical uses copper has no practical substitute. For other purposes, such as tubing, copper clad steel is already coming into wide use with resulting large economies of this material.

MAGNESIUM—Little prospect of change in magnesium restrictions can be anticipated. Desirable as it may be, however, it certainly cannot be considered essential for most civilian needs. Even though it enhances the physical properties of zinc and aluminum die castings, the fact that these materials are themselves under rigid restrictions reduces the hardship caused by the lack of adequate magnesium supplies.

MANGANESE—No adequate substitute for ferromanganese has ever been developed. Essential as this material is in steel manufacture and particularly armor plate, little has ever been produced in this country despite our widespread deposits of low grade ores.

Shipments from Russia, which for years has been our leading source of supply, have practically stopped. In 1940 the Gold Coast, South Africa, Brazil and India each supplied more than was received from Russia, with Cuba shipping somewhat less than the latter.

Deposits Being Developed

Steps are being taken to develop further the Cuban deposits. Combined with the suggested lowering of government specifications for ore to stimulate domestic production this should do much to alleviate the shortage.

MERCURY—Like manganese, many low grade deposits of this ore are available in this country. An inevitable price rise will make profitable the working of these deposits, insuring an adequate supply even if imports are cut off entirely.

NICKEL—Again as in the case of manganese, the most vital use of nickel is as an alloying element for steel and cast iron. Most of our nickel comes from Canada so that an uninterrupted, if somewhat inadequate, supply is assured.

Efforts to use copper instead of nickel in cast iron have met with notable success. For such uses as engine blocks and heads these copper-iron alloys are proving satisfactory. For steels, particularly steel for gears, the solution is not so simple. Molybdenum steels, both carbon and chrome, seem to afford the best possibilities although, notwithstanding the scarcity, some attention is being given manganese.

Neoprene Output Expanding

RUBBER & NEOPRENE—Despite the fact that the major supply source for rubber is the East Indies, such huge reserve supplies have been built up in this country that a serious shortage is remote. Conservation of these stocks is being actively promoted by developing applications for reclaimed rubber which, for many uses, is completely satisfactory. Also, for many consumer goods, so-called hard

(Continued on Page 136)

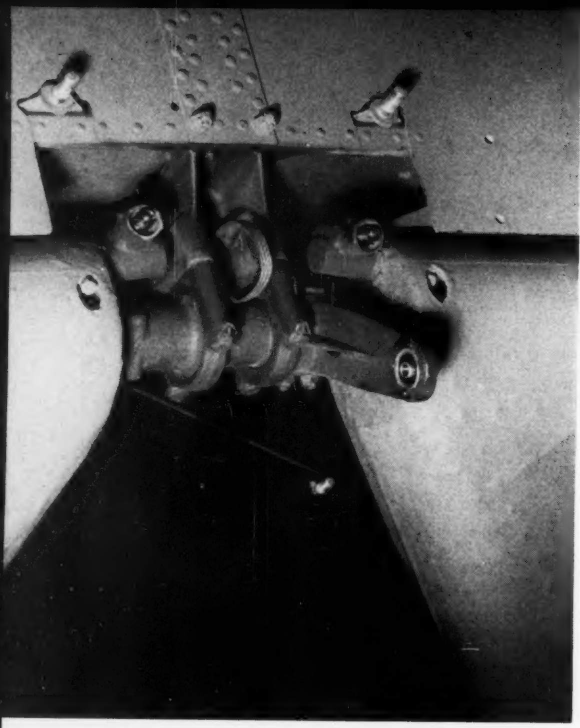


Fig. 1—Typical assembly of ball bearings in aircraft forgings. Control systems require antifriction bearings to assure aerodynamic balance and ease of operation

Aircraft Bearin

By John W. Greve

EMPHASIS on light weight and minimum space requirements in aircraft has led to the development of new series of antifriction bearings. Requirements for aircraft control bearings are consistently similar in one respect—they must have high static load ratings. In addition, to facilitate assembling units such as rudder, elevator and aileron, many ball bearings of the control series are built with extended inner races, providing automatic clearance when installed in links, levers, clevises, etc., as shown in *Figs.* 1 and 2.

Weight reduction and high brinell load have been obtained by reducing the diameter and increasing the number of balls. The first results in a market reduction of the ratio of housing to bore diameters; the second contributes to high static load capacity.

Variations in the proportions of these bearings have, in some cases, led to their specification for such non-aircraft applications as pulleys and cranks without due regard to ratings. This is not recommended, primarily because control series ball bearings are not designed to run at speeds in excess of about 250 revolutions per minute. At higher speeds, they are likely to become noisy. Also, defense requirements have created an acute shortage which bearing manufacturers are bending every effort to elleviate. Appreciating this situation, designers should exercise discretion before contributing to the existing scarcity.

Through hardening, high-chrome SAE 52100 steel contributes to the lightness of the bearing permitting thinner raceways. The inherent properties of this material provide corrosion resistance whereas, in other bearings, cadmium plating is used for this purpose. Complying with Army Air Corps specifications, tolerances are held closely. For example, bore tolerances are only .0005-inch in contrast to the more lenient commercial bearing tolerances.

Roller bearings applied to landing wheels shown in *Fig.* 3 have two primary requirements—load-carrying capacity plus durability and lightness. Load characteristics involve special problems. When flying, these bearings are under practically no load; when aground, the load is purely static. However, when landing or when passing over obstructions in the field heavy impact loads are imposed. An additional factor, rigidity, is essential to assure positive braking.

Fig. 2—Top—Standard K4A is a general utility aircraft bearing. Assembly is usually by press

Fig. 3—Above—Space and weight is conserved in landing wheel assembly by use of a smaller outboard bearing

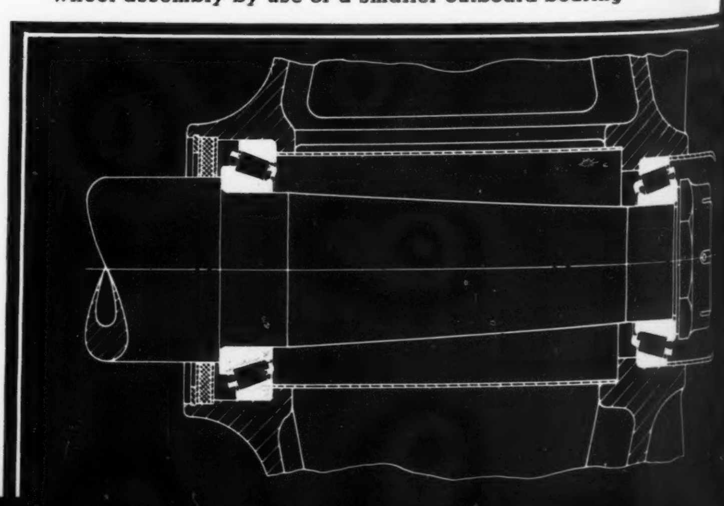


Fig. 4—Right—Method of assembling roller bearing equipped landing wheel. Inboard end of tubular spindle carries most of the load

ringcrease Reliability

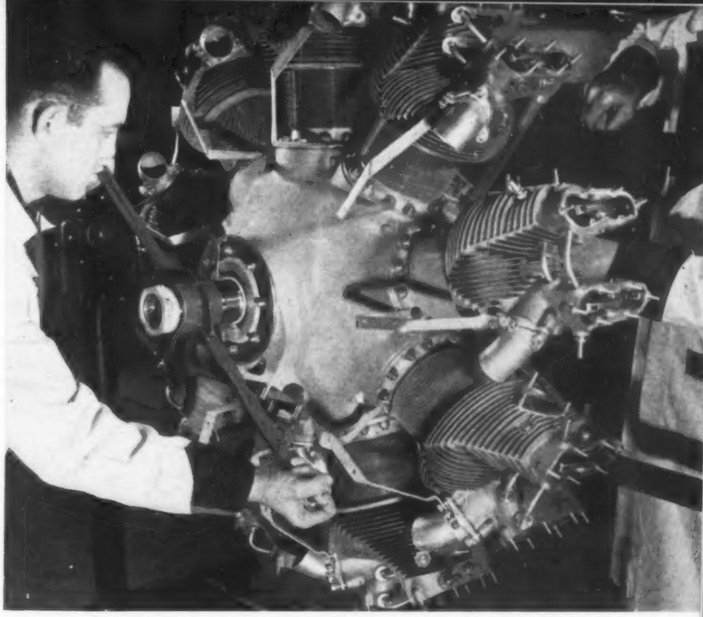
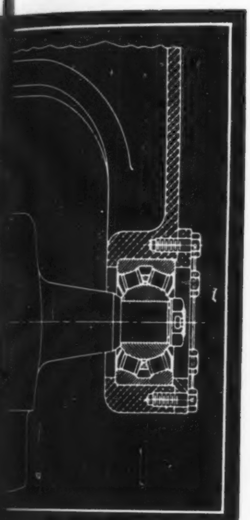


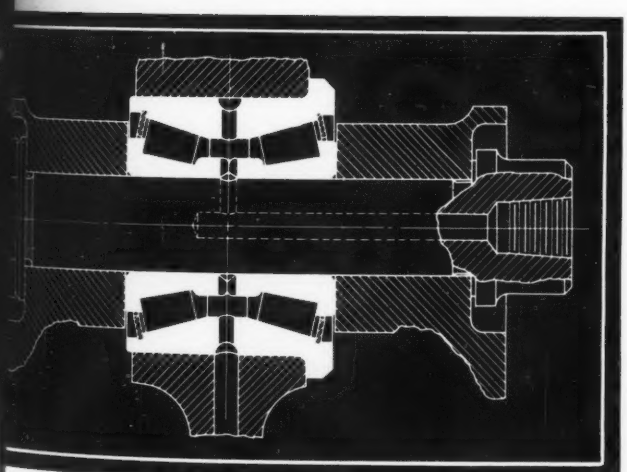
Fig. 8—Buda-Guiberson radial diesel engine used in army tanks has roller bearing equipped rocker arms. Fig. 6 indicates a method of assembly



Since these requirements are nearly constant for all planes it has been possible to develop a practically universal form of wheel mounting as shown in *Fig.* 4. Outer races are pressed directly into the wheel hub and the inner races or cones are given a sliding fit on the spindle. The nut for setting up the bearings is locked with a cotter pin.

Reduced diameter of the end of the spindle permit the use of smaller bearings at the outer ends, taking advantage of lower loads at this point and thus materially reducing weight. Closures are mounted directly in the hub and are held in place with snap rings.

Roller bearing landing wheel assembly of another type is shown in *Fig.* 5. In these bearings the inner race is a spherical element enabling it to move 10 degrees in any direction without altering the contact of the rollers. The curved rollers running on curved races are automatically positioned correctly at all times. Curvatures are so proportioned that substantially live contact is provided under normal loads; with excessive loads, full contact is attained.



Typical of the use of roller bearing in engine rocker arms is the application to an Army tank engine shown in *Fig.* 8. One method of assembly is illustrated in *Fig.* 6. Such an installation provides for radial as well as thrust loads imposed by the angularity of the push rods. Accurate grinding as well as notching of the inner races not only assures positively adjusted preloading but also enables lubricant to be introduced through the rocker arm pin.

Another important application is in the dynafocal link of the engine mounting. In this system of suspension, despite the fact that the supports are in a vertical plane behind the engine, an equivalent to center-of-gravity suspension is achieved. The cylindrical rubber mounting is shown in *Fig.* 7.

For extremely high static loads and for installa-

ft

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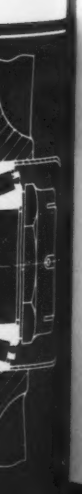
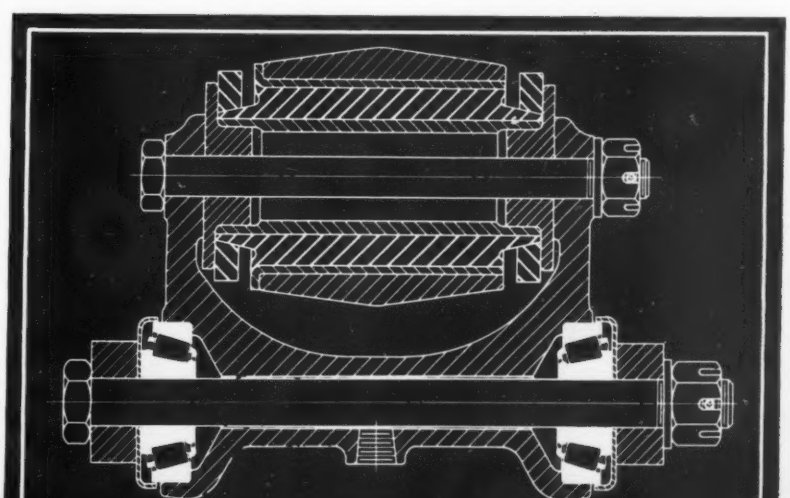
Fig. 5—Top—Concave type roller bearings in landing wheel assembly provide live support of load as well as self-alignment

Fig. 6—Above—Rocker arm bearing assembly indicates lubrication method and manner of preloading

Fig. 7—Right—Bearing assembly for dynafocal system of engine support prevents transmission of engine vibration to the plane





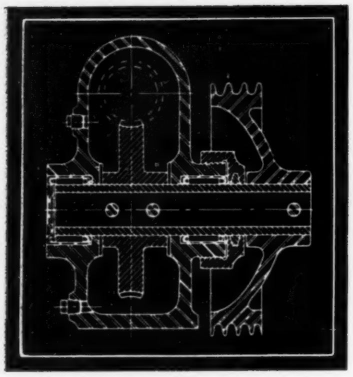


Fig. 9—Bearings in wing flap control gear-box carry high radial loads

tions where space is limited, needle-type bearings have extensive use. A typical application is in the hydraulic actuating cylinder, *Fig.* 11, which operates a retractable landing gear mechanism.

Having substantially the same static friction as running friction, these bearings reliably carry high radial loads with a minimum of loss. For thrust-load applications, however, or combinations of both thrust and radial, ball or roller bearings are used.

In applications such as aircraft landing gear, weight and space may be conserved by using the rotating member as the raceway. Needle bearings for this purpose are available without either inner or outer race or with just the needles alone. In utilizing bearings in this manner, extreme care must be used in following manufacturers' recommendations as to both width and diameter of the races. Finish and hardness are also vital. A through-hardening steel such as SAE 52100 hardened to 60 rockwell *C* is essential for full load capacity. If the hardness is dropped to even 50 rockwell *C* capacity will be reduced 50 per cent.

Some means should always be designed into the assembly for holding the needle bearing components in place. Bronze retainer rings have proved successful in this service. These rings are formed with an annular groove on one face in which tenons on the ends of the needles are guided. In general machine applications, a soft metal is not usually used to take the needle side thrust.

Illustrated in *Fig.* 9 is the use of needle bearings in the gearbox assembly for wing flap controls.

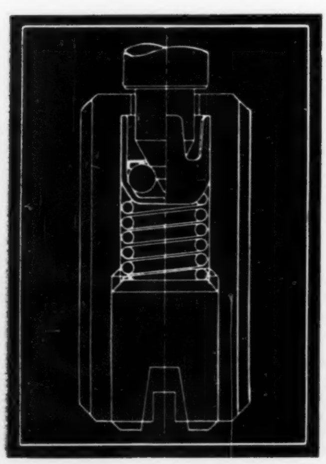


Fig. 10—Instrument ball bearing, loaded by spring, compensates for float and wear of shaft

Because of the low ratio of outside to inside diameters as well as high load capacity, weight of the assembly is minimized.

No less important than proper functioning of operating mechanisms of an airplane is the accuracy of its instruments. Besides precision, there are certain other factors peculiar to aircraft instrument applications. These include rapid changes in temperatures, shock loading, restricted lubrication, necessity for lightness and lack of bulk, as well as inherently trouble-free operation.

Miniature ball bearings are now available in five shaft sizes ranging from .04-inch to 3/32-inch. Corresponding outside diameters range from ⅛-inch to 5/16-inch. What is more important, tolerances are held closer than S. A. E. standards.

The smallest of these bearings has a radial load rating of .2-pound at 20,000 revolutions per minute; the largest, 1.4 pounds at the same speed. At low speeds, the loads are proportionately higher. Such

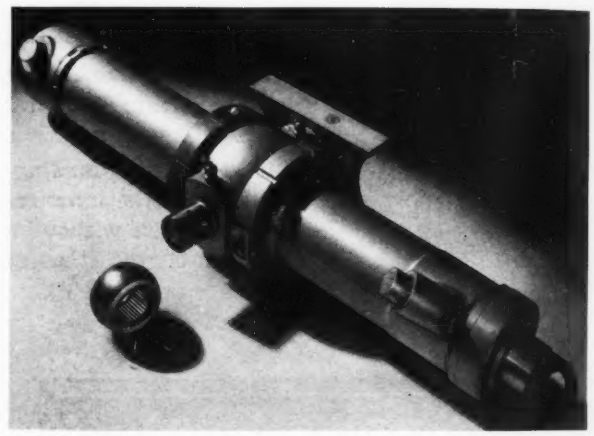


Fig. 11—Needle bearings lend compactness to retractable landing gear hydraulic cylinder assembly

bearings have operated satisfactorily at speeds of 75,000 revolutions per minute. Thrust loads equal to half the applied radial load are permissible. In all design features including materials and tolerances these bearings are the miniature counterparts of the larger precision bearings.

Shown in *Fig.* 10 is a miniature instrument ball bearing of a different type. Designed specifically to replace jewels in pin mountings, the bearings run directly against the hardened pin end tapered to an included angle of 60 degrees. In this particular application the bearing is given a sliding fit in the housing. A light compression spring behind the bearing compensates for wear.

MACHINE DESIGN acknowledges with appreciation the assistance of the following companies in the preparation of this article: Curtiss-Wright Corp., Fafnir Bearing Co., Miniature Ball Bearing Co., Miniature Precision Bearings Co., New Departure Co., Nice Ball Bearing Co., Norma-Hoffmann Bearings Corp., Roller Bearing Co. of America, Shafer Bearing Corp., Timken Roller Bearing Co., The Torrington Co.

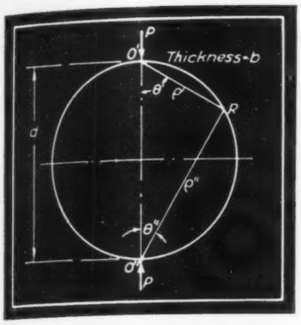
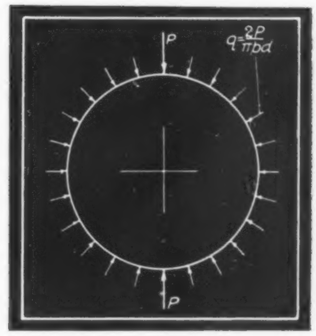


Fig. 64 — Left — Diametral loading of a disk or long roller

Fig. 65 — Right — Load producing simple stress systems which are radial from each load



Applying Theory of Elasticity in Practical Design

By R. E. Orton
Chief Engineer
Acme Steel Co., Chicago

Part VII

Loads on Rollers and Wedges

USEFUL to indicate the safety against rupture of rolls in roller bearings, mill rolls, bridge rollers and other types of solid rollers, the solution in this section considers stress distribution in a solid disk, or long roller, under two or more concentrated loads. Its derivation is an extension of the stresses produced in a semi-infinite plate by a load applied normally to its boundary as covered in the preceding section.

Attention is again directed, therefore, to the theoretical nature of a concentrated load. The stresses immediately adjacent to the load will depend upon the local elastic deformation produced by the contacting body. This "skin" stress is given by the Hertz equations, to be discussed next month. Frequently rollers will be case hardened, the case thickness and strength being such as to resist these skin stresses. The core then, is designed on the basis of the solution given here.

With two equal and diametrally opposite concentrated loads, as shown in *Fig.* 64, it may be assumed that each load produces its own stress system and that the systems are given by the solution for the semi-infinite plate. Extension of these two systems to the boundary will then determine what loading, in addition to the two concentrated loads, is needed to maintain the assumed distribution. The addition now of a loading equal in value but opposite in direction to the loading just determined, will free the boundary of all loads except the two concentrated loads. Combination of the stress system produced by this "cancelling" loading with the two stress systems assumed for the concentrated loads, will give the distribution from the concentrated loads alone.

The process outlined above is another case of superposition of loads with their stress systems. Putting it another way, the procedure is to determine the stress system from the loadings shown in *Figs.* 65 and 66 and then superimpose adding the loads and stresses. The distributed load, being equal in value but opposite in direction, cancels

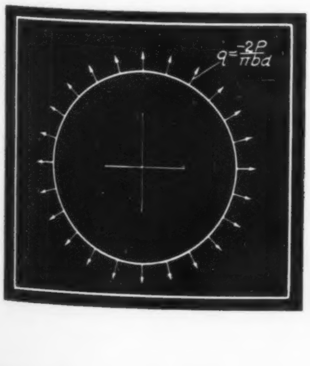
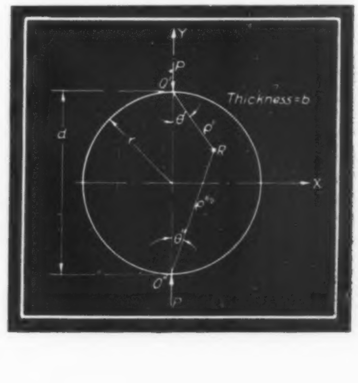


Fig. 66—Left—Loading produces uniform hydrostatic tension. Combined with the loading in Fig. 65, this gives the distribution for the loading of Fig. 64

Fig. 67 — Right — Stress within disk is a combination of simple radial stress from O' and O'' together with a hydrostatic tension from Fig. 66



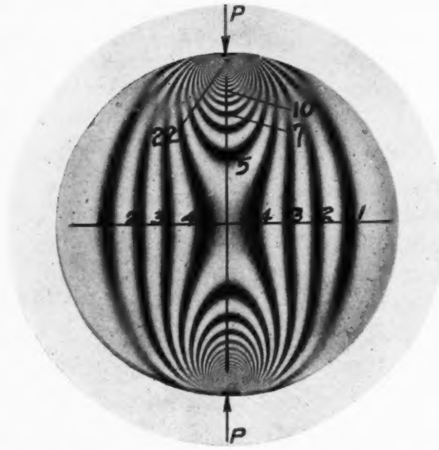


Fig. 68—Adjacent—Photoelastic picture of a circular disk under diametrally opposite cencentrated loads. Enlargement $1\frac{3}{4}$ diameters

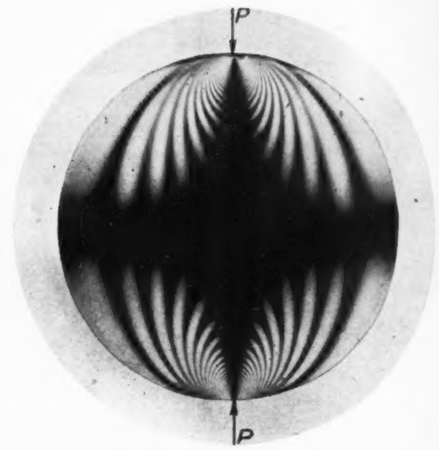


Fig. 69—Right side—Isoclinic picture of disk in Fig. 68. Polarizer is parallel to loads. Enlargement $1\frac{3}{4}$ diameters

out leaving only the concentrated loads.

As stated, it is assumed that each of the concentrated loads shown in *Fig.* 64 produces the simple radial stress distribution given by Equations 105 and 106 of the preceding section. With this distribution, the principal stresses are radial and tangential and the tangential stress is zero. The angle $O'RO''$, at any point R on the boundary of the disk, subtends a diameter and is therefore a right angle. The two stress systems, then, have coincident principal directions, with the S_1 stresses 90 degrees apart. Their combination, therefore, will have the

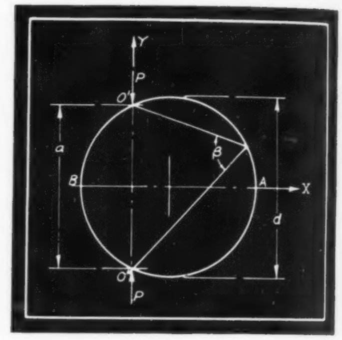


Fig. 72 — Right — Solution for loads applied along a chord follows method of diametral loadings

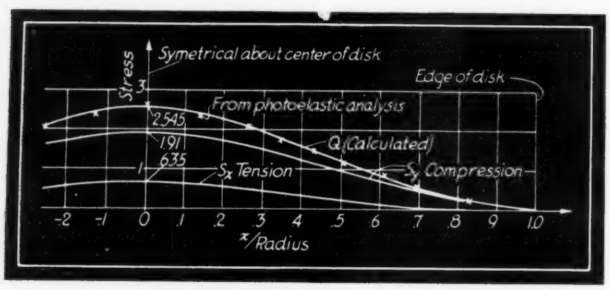


Fig. 70—Stress across diameter of disk perpendicular to loading. $P=b=d=1$. For other values, multiply by P and divide by bd. Here $v_{xy}=0$

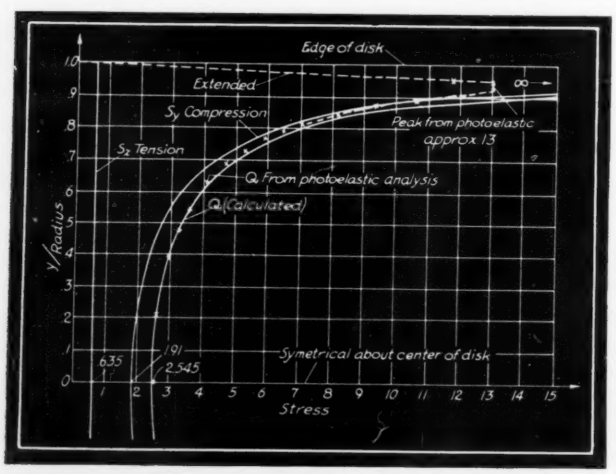


Fig. 71—Stress along diameter along which loads are applied. Values and ruling are the same as given for Fig. 70

same directions and will be given by the algebraic sum. Equation 107 from the preceding section gives the same value of the stress from either of the loads, g having the value d. The resultant stress along the boundary is a hydrostatic head given by

$$S_1 = S_2 = \frac{-2P}{\pi bd}$$

To maintain this stress will require a normal distributed loading of constant value, given by

$$q = \frac{2P}{\pi bd}$$

as shown in *Fig.* 65.

To the loading of Fig. 65 is now added the normal loading

$$q = \frac{-2P}{\pi bd}$$

as shown in *Fig.* 66. The stress distribution from this last loading is a uniform hydrostatic tension equal to $2P/\pi bd$, as given by Equation 71 in Part IV (May issue).

The stress at any point R within the disk, *Fig.* 67, is obtained by combining the following three stress systems:

System 1. Simple radial from O',
$$S'_\rho = -2P \cos\theta' / \pi b\rho'$$
System 2. Simple radial from O'',
$$S''_\rho = -2P \cos\theta'' / \pi b\rho''$$
System 3. Hydrostatic tension,
$$2P/\pi bd.$$

This combination could be made by means of Mohr's circle diagrams as has been done in several other cases. In this case, however, it is simpler to reduce each system to rectangular co-ordinates and then to add algebraically. The origin of co-ordinates will be taken at the center of the disk with directions parallel and perpendicular to the load direction, as shown in *Fig.* 67.

System 1 is already given in rectangular co-ordinates by Equations 109, 110 and 111 of the preceding section, except that the origin is given there at O'. Transferring the origin to the center of the disk these equations become

$$S_x = \frac{-2Px^2(r-y)}{\pi b(\rho')^4}$$

$$S_y = \frac{-2P(r-y)^3}{\pi b(\rho')^4}$$

$$v_{xy} = \frac{2Px(r-y)^2}{\pi b(\rho')^4}$$

System 2 is also obtained from Equations 109, 110 and 111. The origin is now at O'' and P has opposite sign. Transferring the origin to the center

$$S_x = \frac{-2Px^2(r+y)}{\pi b(\rho'')^4}$$

$$S_y = \frac{-2P(r+y)^3}{\pi b(\rho'')^4}$$

$$v_{xy} = \frac{-2Px(r+y)^2}{\pi b(\rho'')^4}$$

Values ρ' and ρ'' in these equations are given by

$$\rho' = \sqrt{(r-y)^2+x^2}$$

$$\rho'' = \sqrt{(r+y)^2+x^2}$$

System 3, in rectangular co-ordinates, is given by

$$S_x = S_y = \frac{2P}{\pi bd}$$

$$v_{xy} = 0$$

Adding these three sets of equations and making the substitutions $x_1 = x/r$, $y_1 = y/r$, $\rho_1 = \rho'/r$ and $\rho_2 = \rho''/r$ gives finally for the resultant stress distribution[1]

$$S_x = \frac{2P}{\pi bd}\left[1 - \frac{2x_1^2(1-y_1)}{\rho_1^4} - \frac{2x_1^2(1+y_1)}{\rho_2^4}\right] \quad (116)$$

[1] Original solution of these disk problems is credited to H. Hertz, 1883.

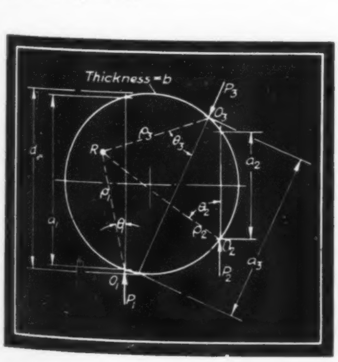


Fig. 73 — Left — Random loading in static equilibrium. Simple radial stresses are combined by Mohr's circle with hydrostatic tension to obtain solution

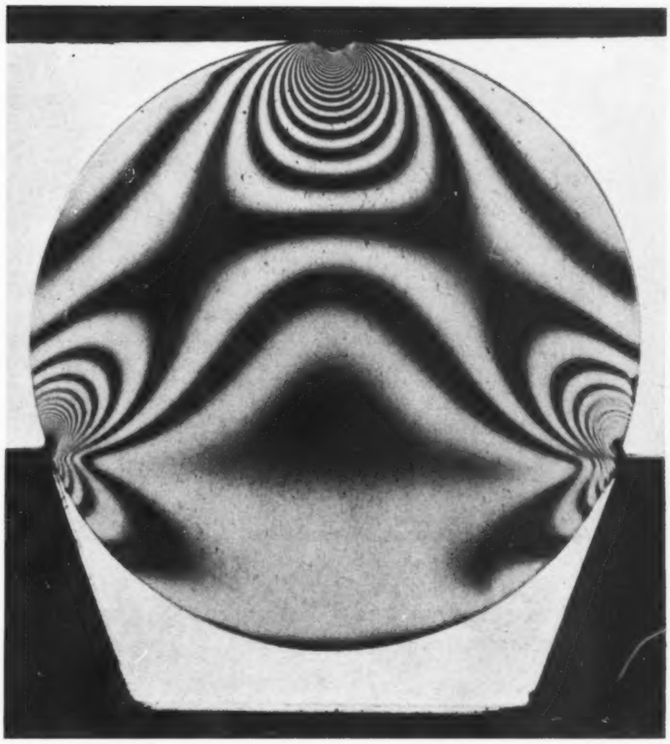


Fig. 74—Above—Light field photoelastic picture of disk with three loads

$$S_y = \frac{-2P}{\pi bd}\left[\frac{2(1-y_1)^3}{\rho_1^4} + \frac{2(1+y_1)^3}{\rho_2^4} - 1\right] \ldots (117)$$

$$v_{xy} = \frac{4Px_1}{\pi bd}\left[\frac{(1-y_1)^2}{\rho_1^4} - \frac{(1+y_1)^2}{\rho_2^4}\right] \ldots\ldots (118)$$

S_x is always tension, S_y always compression. All three stresses along the boundary, except at the load points, are zero. Therefore the stress in all directions at the boundary is zero.

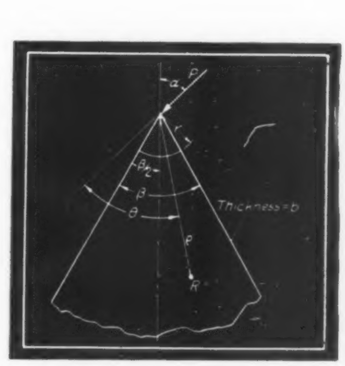
The shear stress v_{xy} is zero along both the X and Y axes; principal directions, therefore, are parallel to the axes along these two diameters. Along the horizontal diameter Equations 116 and 117 reduce to

$$S_x = \frac{2P}{\pi bd}\left(\frac{1-x_1^2}{1+x_1^2}\right)^2 \ldots\ldots (119)$$

$$S_y = \frac{-2P}{\pi bd}\left[\frac{(3+x_1^2)(1-x_1^2)}{(1+x_1^2)^2}\right] \ldots\ldots (120)$$

Along the vertical diameter they become

Fig. 75 — Right — Inclined concentrated load on the apex of an infinite wedge



$$S_x = \frac{2P}{\pi bd} \quad \text{(121)}$$

$$S_y = \frac{-2P}{\pi bd}\left(\frac{3+y_1^2}{1-y_1^2}\right) \quad \text{(122)}$$

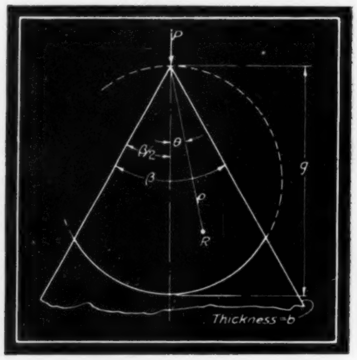


Fig. 76 — Left — Symmetrically loaded wedge. Stress along circle g is constant

The principal stress difference, *Q*, along these two diameters is given by

$$Q = \frac{8P(1-x_1^2)}{\pi bd(1+x_1^2)^2}, \qquad y_1 = 0 \quad \text{(123)}$$

$$Q = \frac{8P}{\pi bd(1-y_1^2)}, \qquad x_1 = 0 \quad \text{(124)}$$

At the center all three stress equations are

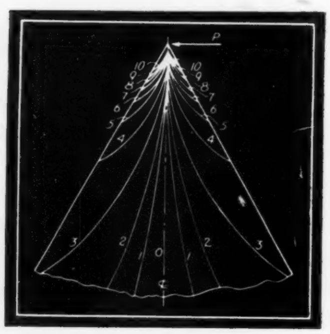


Fig. 77 — Left — Photoelastic fringe pattern for horizontally loaded wedge

$$S_x = \frac{2P}{\pi bd} \quad \text{(125)}$$

$$S_y = \frac{-6P}{\pi bd} \quad \text{(126)}$$

$$Q = \frac{8P}{\pi bd} \quad \text{(127)}$$

In *Fig.* 68 is shown a photoelastic fringe picture of a disk under two diametrally opposite concentrated loads. *Fig.* 69 is an isoclinic of the same disk, with polarizing axis parallel to the load direction. The parallel and perpendicular isoclinic bands in the latter picture confirm the conclusion that the principal directions are parallel and perpendicular to the load direction along these two diameters.

Stress distribution along the horizontal diameter is shown in *Fig.* 70 and in *Fig.* 71 is seen that along the vertical. The *Q* stress obtained from the photoelastic analysis is also plotted. In obtaining this latter curve a light field picture and an 8¼-diameter enlarged view of the portion around the load was used in conjunction with *Fig.* 68. Load of 160 pounds was applied to the 1.085-inch diameter bakelite disk by means of two parallel steel plates. The model thickness is .276 inch and $H = 86.6$.

Photoelastic results plotted in *Figs.* 70 and 71 show remarkable agreement with theory. Accuracy is here estimated at ± 5 per cent except in the region near the load.

The photoelastic result, which is presumed to be the actual stress existing in the piece, should drop away from the theoretical curve near the load, as it does. This is the region where recourse must be made to the Hertz equations. The portion of the photoelastic curve is also somewhat misleading as it is not correct to convert to a "unit disk and load" by simple multiplication and division, as the variation is not of the first power. This will be discussed later in detail under contact stresses.

In *Fig.* 68 the ring fringe pattern is characteristic of a contact load. Near the load the fringes closely approximate a circular shape, although the circles are not quite tangent to the boundary because the load is actually distributed over a small width. As the center of the disk is approached the influence of the other load is felt and the circles become more and more elongated, finally merging into the composite pattern.

Solution for opposing loads applied along a chord of the disk, *Fig.* 72, follows the pattern of the preceding solution. The hydrostatic tension to be added to the system radial from O' and O'' is given by

$$S = \frac{2P \sin \beta}{\pi bd} = \frac{2Pa}{\pi bd^2} \quad \text{(128)}$$

where β is the angle inscribed in the circle and subtending the chord along which the loads are applied. The $\sin \beta$ is constant regardless of the location along the boundary of the vertex of the angle β, and is given by $\sin \beta = a/d$, where a is the length of the chord.

Axes location for this problem is again along and perpendicular to the loads, as shown in *Fig.* 72. The transfer of origins from Equations 109, 110 and 111 is from O' and O'', so that the quantity r in the earlier equations is replaced by $a/2$.

As before, the principal directions are parallel and perpendicular to the loads along the X and Y axes. The stress distribution is somewhat similar to the diametral loading, the closeness of the agreement depending upon the eccentricity of the chord. Along the X axis S_x is zero at A and B; S_y is tension at A, zero at some intermediate value of x, and compression at B. Q likewise will differ from zero at A and B, and be zero at $x = a/2$. S_x along the Y axis is constant and equal to $2Pa/\pi bd^2$.

(Continued on Page 128)

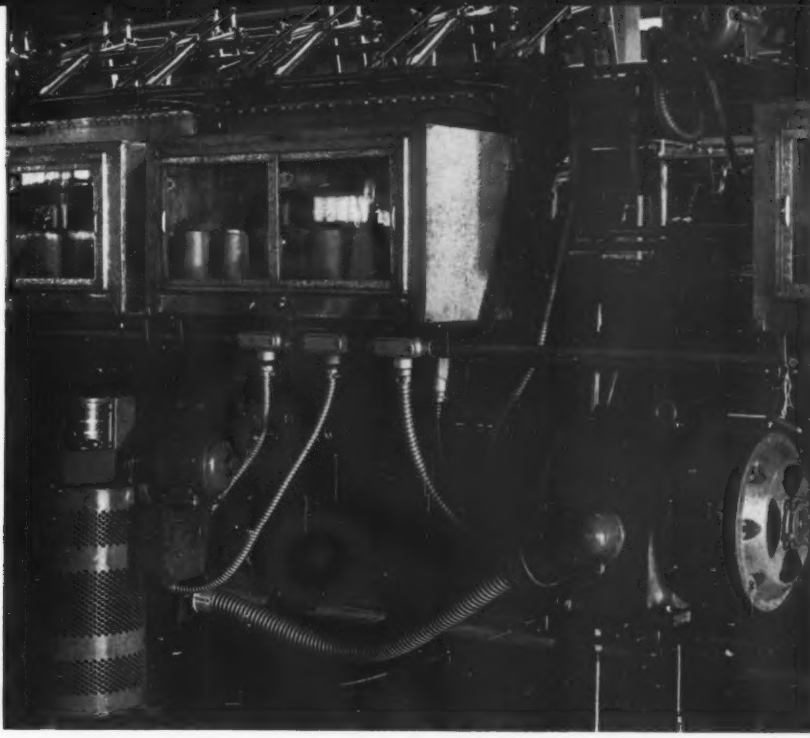


Fig. 1 — Adjustable - speed motor driving a knitting machine. Preset head is located on top of regulator at left for automatic speed changing

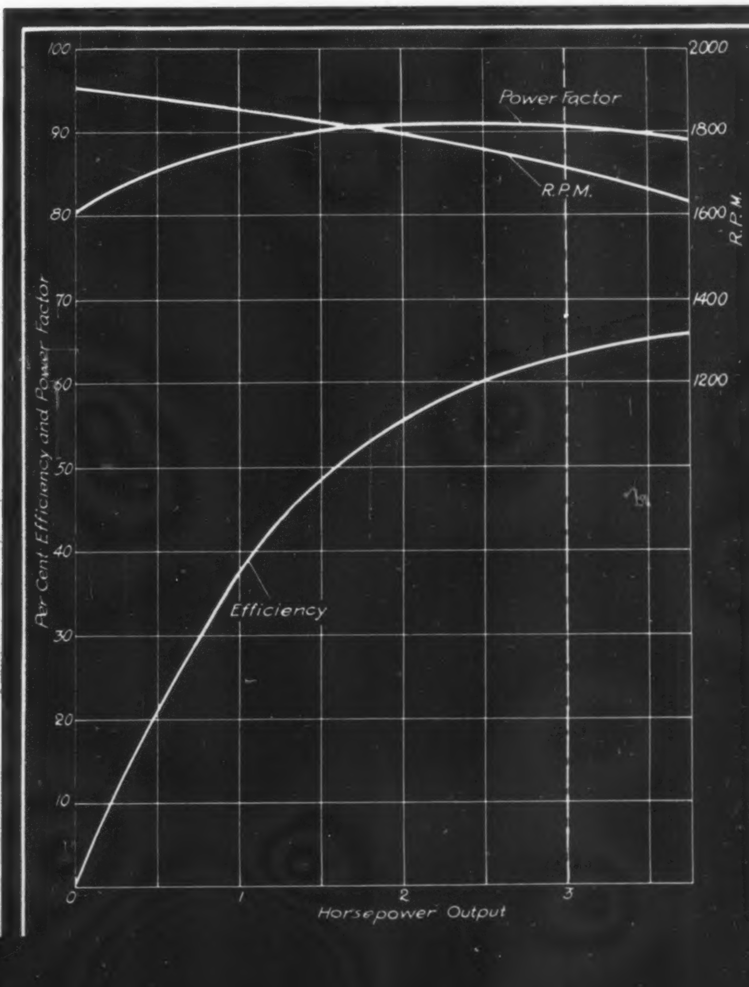
Selecting Special Motors

Part VIII—Adjustable Speed Motors

By H. M. Edmunds

Design Engineer
Crocker-Wheeler Electric Mfg. Co.

Fig. 2—Performance curves for a 3-horsepower adjustable-speed motor at full speed. Motor is rated 3/1 horsepower, 1720/580 revolutions per minute, 220 volts, 60 cycles, 3 phase



MACHINE drives in which a speed range is furnished by motor and control employ motors which are classified either as *adjustable* or *variable* according to the closeness of speed regulation which may be met at any setting of the regulator. If this regulation must be close, *adjustable* motors are required. If a wide variation is permissible, power units classed as *variable* may be utilized.

Direct-current shunt motors are essentially in the adjustable class, whereas the slip-ring or wound-rotor motors are variable. To the adjustable class belong most machine tool drives where cutting speeds should remain steady, even with interrupted cuts, while in the variable class are centrifugal fans, pumps and drives which make a fixed torque demand at any given speed.

In the adjustable speed drives no dependence is made on the load to remain steady at any given speed. Such drives approach the theoretical ideal. Some forty years ago the Ward Leonard type of drive was developed which laid-down the basic

principles of the adjustable speed system. These are briefly as follows:

1. Power motor is a direct-current shunt motor run at full excitation through speed range
2. Armature is fed with an adjustable voltage fixed for any one speed setting. This voltage is furnished by a motor-generator set, with the adjustments made by rheostatic control of the field of the generator. By this method the power-carrying circuit of the generator and motor armature is never interrupted even at reversal
3. Motor-generator set must have an exciter to furnish field current for generator and motor
4. So-called "suicide" connection removes the electromotive force generated by residual magnetism in the generator when the field circuit is opened. This is done by a temporary connection between the generator field and the motor armature.

By the Ward Leonard system, a speed range of 8 to 1 can be obtained and a further 2 to 1 reduction can be given at the top end by field weakening, making a total working range of 16 to 1. Driving motor may be either squirrel cage or synchronous. The latter gives slightly better regulation because there is no dip in speed at high load.

In some applications the condition of good regulation to the low end of the speed range to prevent stalling at high torques requires compounding of the generator. This condition may be the determining factor in selecting the size machine to be used.

An example in a machine tool application shows how the low-speed regulation may determine the size of power unit to be used. A group of lathes for accurate machining of optical lens mounts could be handled with a 1-horsepower motor except that one particular operation called for slow speed free running of a chaser for cutting a small thread. A motor speed of 120 revolutions per minute was the highest at which the impact of engagement could be permitted and at this speed with the 1-horsepower motor the load of operation stalled the motor. A motor of three times the weight and capacity was necessary to overcome this difficulty.

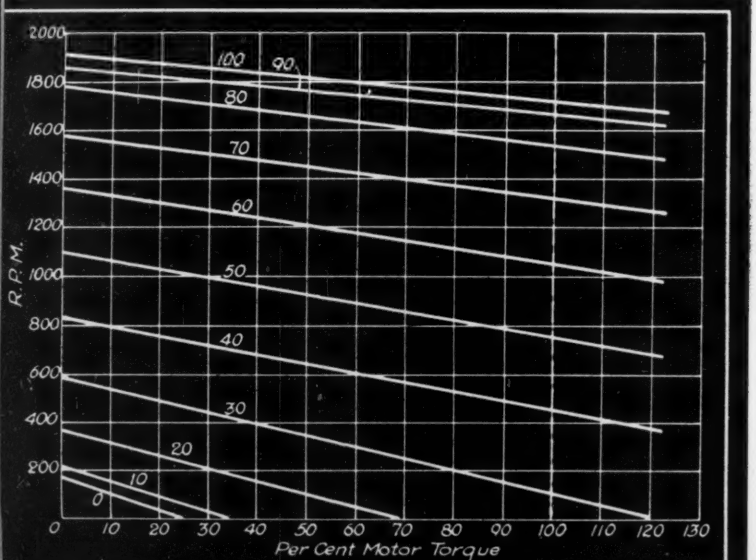


Fig. 4—Speed-torque curves of a 3-horsepower polyspeed motor with respect to various regulator dial settings. Motor has same rating as shown in Fig. 2

As an alternate to Ward Leonard sets, a combination has recently been developed for alternating-current circuits which uses an induction motor and direct-current series generator instead of the usual set. The direct-current generator and the motor driven by it are both series wound and speed regulation is obtained by adjustment of the generator field strength by a rheostat in shunt with it. The speed-torque curves shown in *Fig.* 3 are remarkably flat at low speeds. *Fig.* 2 shows speed regulation obtainable.

Speeds May Be Preset

There are many points of similarity between the Ward Leonard drive and the use of alternating-current commutator or polyspeed motor with induction regulator control.

Both are constant torque drives which means in effect that a motor say of 10 horsepower at 600 revolutions per minute will have a capacity of 30 horsepower at 1800 revolutions per minute. Therefore the active material of the motor is worked to its full capacity at all speeds within the speed range. Control for the polyspeed motor is in the form of a drum in which the primary is movable and the secondary stationary. As with Ward Leonard systems the power circuit, consisting of the motor armature and the regulator secondary, is never interrupted. Control of the voltage applied to the motor brushes is made by relative movement of the regulator changing the mutual flux linkages between primary and secondary. Speed adjustment is thus stepless. Movement does not involve rubbing of contacts while current is passing and for this reason the system is adaptable to pilot motor control with preset head with a number of sections which can be set for any speed.

This feature has been much used in the textile industry, particularly for knitting machines such as shown in *Fig.* 1, in which the various phases of narrowing, welt turning, etc., can be accomplished at the best speed for product quality. Other uses include printing presses, speed control of power-operated ladles in foundries, speed testing of grinding wheels, and is now used for testing fuses.

Where the conditions of the load admit of its use, the slip-ring or wound rotor motor has many advantages on the score of economy of space and cost over the Ward Leonard group or the alternating-current commutator motor. The absence of a commutator is also an advantage.

Though essentially in the variable-speed class, its selection often is advisable for many borderline cases. Slip-ring motors are built up to almost any size—from 40,000 horsepower at 300 revolutions per minute used for wind tunnel testing down to the NEMA 224 frame size of three-quarter horsepower at 900 revolutions per minute.

Slip-ring motors are extensively used for news

printing machines, an application which calls for special design to meet the severe conditions of service. The press may be required to operate at top speed to meet emergency demands or more slowly for normal production. A threading speed has to be provided but this is furnished by an auxiliary motor driving the rotor of the main motor through gearing and an overrunning clutch.

Smooth acceleration from threading to working speed and emergency stopping are essential. To reduce the heavy rheostat losses for slow speed running, a two-speed gearbox sometimes is used.

As a guide to the selection of a power unit for speed range drive, the following set of questions should be considered. A certain latitude in the selection may be taken, and it should be borne in mind that adjustable drives cost more than variable and take more space, but from a performance angle close regulation has great advantages.

1. *What horsepower is required?*

In constant-torque drives, horsepower is quoted as at the top of the constant-torque range. Where a constant-horsepower range is tacked on at the top, this will be at quoted horsepower. Fan builders are always conservative in quoting horsepower consumptions of fans. No extra margin need be allowed in such cases.

2. *What speed range is necessary?*

Ranges over 2:1 are generally unsuitable for slip ring drives although they can be furnished for 3:1 in special cases.

3. *How does load vary throughout the range?*

This is important, particularly if slip-ring motors are considered. The slip-ring motor and, in fact, all induction motors can be regarded as equivalent to a slipping clutch—having the same torque in both clutch halves but wasting the energy of that torque times the slip under all conditions.

4. *Is load constant at any one speed, or how may it vary?*

This question is the essence of the problem. If the answer is affirmative, the possible use of slip-ring motor is indicated provided questions (2) and (3) are satisfied by a slip-ring drive. If the load fluctuations at any given speed are wide then equipment with adjustable control must be used.

5. *Is running continuous at any one speed, requiring infrequent adjustment?*

In several industrial drives, such as woodworking machines, grinding of antifriction bearing parts, and others, the use of frequency changing drive of squirrel cage motors is adopted where quantity production justifies the installation. The equipment for giving variable frequency starting from alternating-current supply is complicated. One method is to use a motor-alternator combination with an adjustable pulley drive between.

6. *Is it a program drive following some prearranged cycle of speed variation?*

Knitting machine drive described above is an example of this class of drive. Control calls for special study.

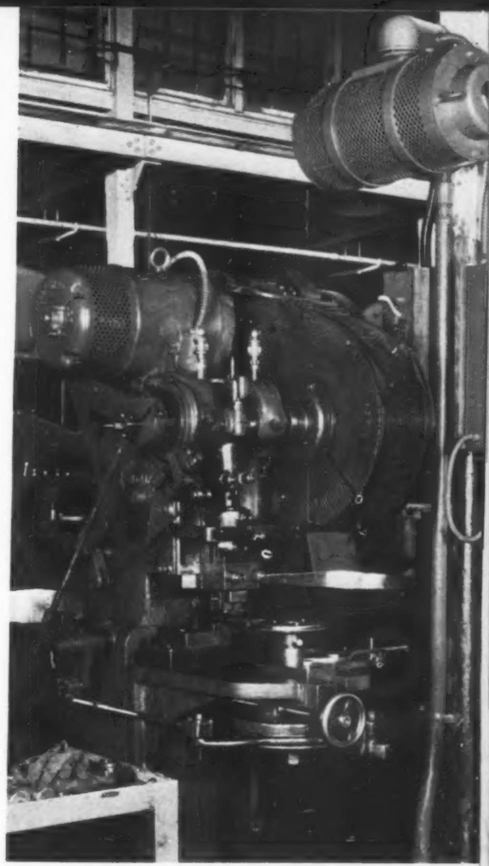


Fig. 3—Punch press utilizes adjustable-speed drive to accommodate varying classes of work

7. *Do speed changes take place while running, and, if so, are there any restrictions as to snatch in changing speed?*

In calendar drives accelerating from threading to running, it is important to avoid snatch.

8. *Is a very slow speed for threading required, or would "inching" or "jogging" be satisfactory?*

For small powers, the adjustable feature can generally take care of threading. In heavy printing press work, small gearmotor and clutch are used.

9. *Is it desirable or essential that speed control should be stepless? If not, what speed interval between steps is admissible?*

Some grinding operations call for stepless control. With vernier rheostat, fine speed adjustment can also be had with the direct-current drives.

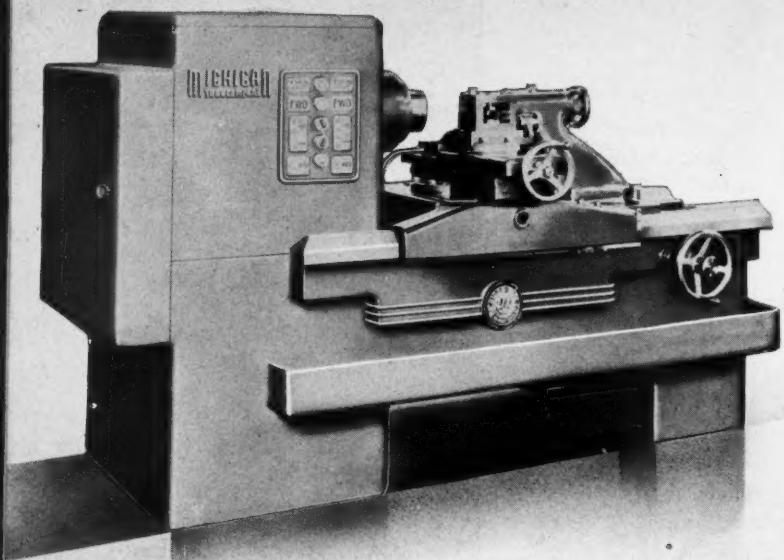
10. *Is emergency stopping required in addition to normal stopping by shutting off or reducing speed to zero by regulation?*

Emergency stopping is usually done by independent or motor mounted brakes. With the polyspeed motor, regenerated power is fed to the line on stopping, providing a beneficial cushioning action.

For light power applications such as small printing presses and mechanisms imitating the motions of a horse or camel, small single-phase repulsion motors are used. The torque, and therefore the speed, are varied by brush shift. Powers do not run much above 1 horsepower at 1500 revolutions per minute. For suitable applications this drive should be considered.

Numerous mechanical variators driven by a constant speed motor do not come within the scope of this article but it should be noted that their possible use should be studied.

One feature of mechanical variators is that the motor is small, of high speed and uses its active material to the best advantage. The torque is increased by the ratio of reduction of the mechanism so that the combination gives constant horsepower.



Left—Two built-in reversible motors coupled in such a manner that, by operating either one or both, seven spindle speeds are obtained typifies Michigan cutter relieving machine. Relieving from any angle is permitted by mounting the cam slide on a turntable. Change from one cutting operation to another is facilitated by quick-interchangeable cams

Right—Yoke-type front head having a built-in punch and die that nibbles out rectangular shavings of metal distinguishes the Thor portable electric nibbler. Capable of cutting steel up to 18 gage and aluminum up to 15 gage, the machine makes its own clearance without distortion or curling of the sheet. Flexibility for cutting curves or angles in either flat sheet or tube is obtained by making the head adjustable at angles of 30 or 90 degrees to either the right or left when working in limited space

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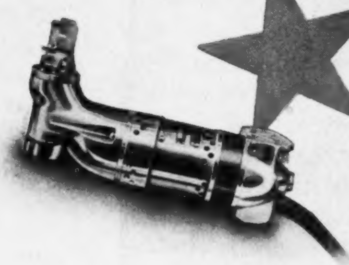

Below—Completely automatic operation of ice cream freezer is notable achievement of mills unit. Freezing time, agitating time, washing control and maintenance of proper temperature after completion of batch are all regulated by adjustable dial indicators mounted on the front of the machine. Forced air circulation by blowers over refrigerating coils reduces hadening time in auxiliary unit by over half without necessitating increased size of condensing unit

THIS MONTH'S COVER— wo units
having 16 stations, per ate mac
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Greenlee transfer mach e of co
engine cylinder head eve s as co
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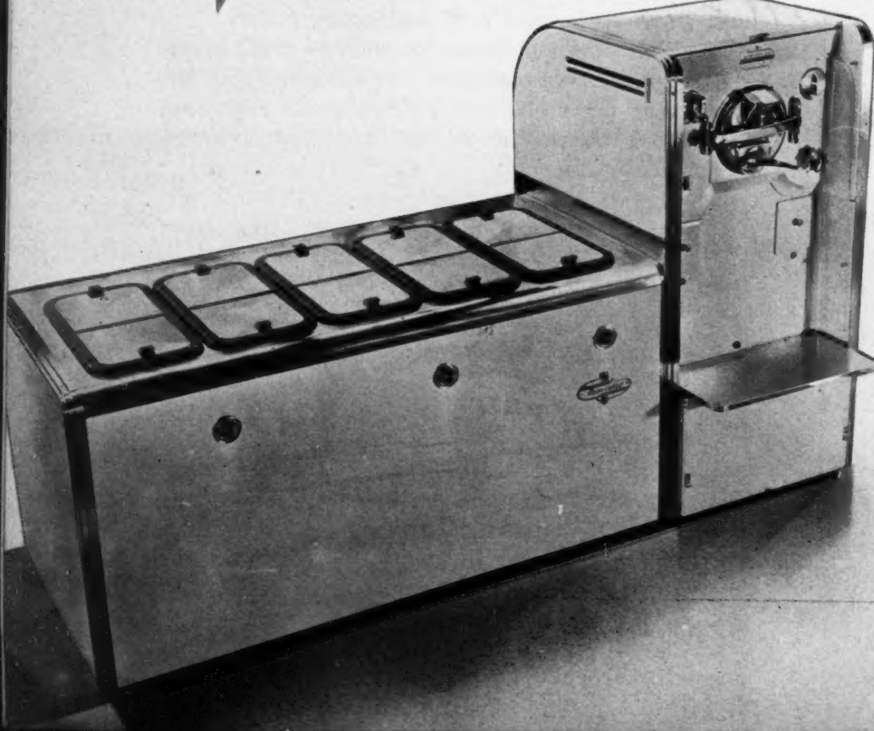
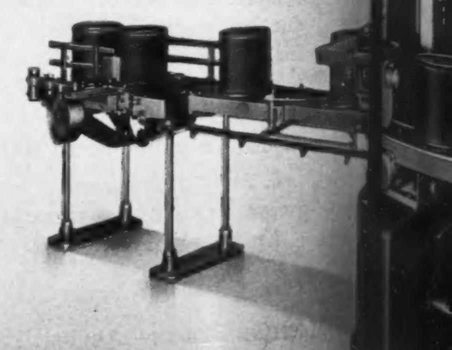
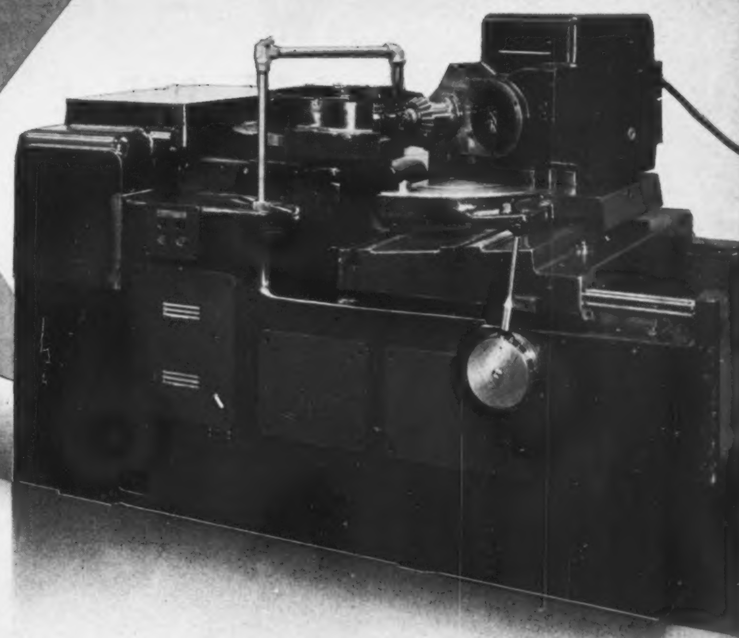
Right—Utilizing a special aluminum alloy having a tensile strength equivalent to that of mild steel, the Cameron can seaming machine is capable of adjustment for four different heights of pails. This obviates the necessity for adjusting the conveyor level. Excessive wear on the seaming head is eliminated by using four arms and rollers in two sets, which also permits maximum processing speeds





Left—Standard motors driving through heavy-duty V-belts provide power for horizontal disk grinder. Advanced design of louvers cast in removable guard ring assures efficient removal of grinding dust and grit. Swinging bar-type dresser arm permits location of ball bearing dresser head at any point on 30-inch diameter of Gardner machine. Combination radial and thrust-type ball bearings rated at 10,400 pounds provides mounting for wheel spindle



Above—Provision for horizontal motion of the cutter spindle combined with the improved design of cutter blades results in proper taper and profile shape of gear tooth slots. Hydraulic chuck and hydraulic drive of work head enhances speed and ease of operation in Gleason straight bevel gear tooth rougher by making possible all operations through a single control lever. Change gears control indexing of spindle

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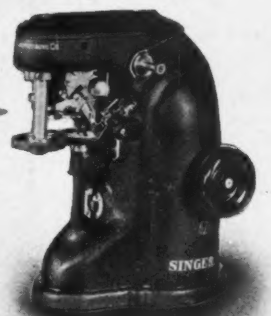


Left—Built-in lighting contributes to ease of threading and guiding of selvages in Singer sewing machine for seaming women's hose. Unobstructed view of work guides and feed cups is provided by greatly reduced width of machine head. Automatic lubrication system and reduced number of moving parts all of which are dynamically balanced assure vibration-free operation with a minimum of maintenance

Right—Individual cylinder heads and oil-tight covers over working parts contribute to cleanlined appearance of eight-cylinder diesel engine. Fuel pumps capable of adjustment by means of cetane selector provides utilization of any type of fuel with maximum efficiency and economy. Special piston crown design combined with improved inlet valves in Clark engine induce air stream turbulence, insuring complete mixing of air and fuel. Thorough dispersion of fuel is achieved by centrally located multi-orifice injection nozzle

Aid to Engineering Departments

NEED for manpower, most obvious on defense production lines, is being felt no less stringently in engineering departments. Few chief engineers can be found who could not utilize to advantage twice as many men as are now available to them.

Under current circumstances most heads of engineering departments have to be satisfied with adding to their staffs young men from colleges and schools, or trainees under the defense program. Such additions, however, call for further training periods and create in the minds of management a wish that more qualified men could be "begged, borrowed or stolen from other sources.

An intriguing suggestion along these lines has been made by the manager of a machine tool company at present swamped under a file of orders for special machines. Tooling is his problem. He feels that if a "flying squad" of tool designers could be formed to descend on a plant, do the tooling up and pass on to the next, the problem would be solved. This seems to be an idea that the American Society of Tool Engineers might well look into.

A similar method, though difficulties would be numerous, could probably be worked out for other divisions of engineering departments. Detailing, for instance, is causing a tieup in many companies and is taking the services of men qualified for more creative work. Could not the national engineering societies, the consulting engineers in key centers and the colleges at which intensive training courses are being held, get together and establish headquarters for drafting work? These could be utilized by machinery companies working under pressure. Such centers would, while specializing in drafting, serve as excellent training grounds for potential designers and thus supplement the efforts of the colleges to meet the increasing demands arising from the defense program.

Selecting Materials

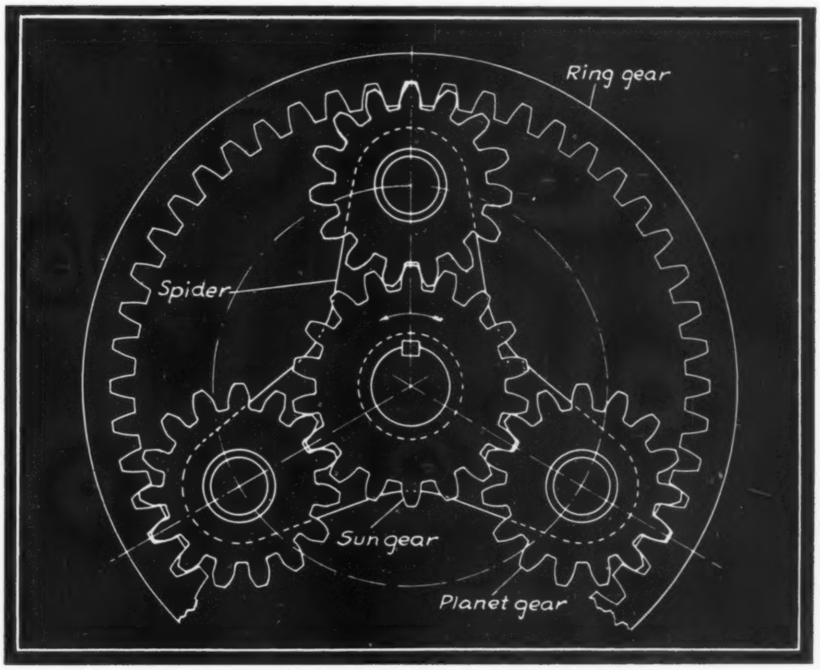
ONE OF the reasons for the Nazi attack on Russia is given as the increasingly urgent need in Germany for raw materials essential to the production of war equipment. Only Hitler knows the complete answer but the fact remains that recognition is definitely being given everywhere to the possibility of the war's being won or lost on the availability of materials in conflicting countries.

To hasten our own defense program, vastly increased facilities for producing basic metals are being developed. Substitutes are being found for the more strategic materials in which shortages occur, and imports—where domestic supplies are lacking—have been sharply increased. But even though these efforts attain a large measure of success it may be years before a stable, let alone a normal, condition is reached.

Designers of machines will need all the aids to selection or substitution of materials that may become available during this time. Not the least of these is MACHINE DESIGN'S Directory of Materials, ninth edition of which will form part of the forthcoming October issue. Every effort is being put forth to adapt the new edition to the current pressing conditions and the difficulties ahead.

Design of Planetary Gear Systems

By George S. Hoell
Edw. G. Budd Mfg. Co.



In planetary gear systems (using standard gears) the ratio and number of planet gears are of controlling importance. The following design data make possible the determination of both the number and location of planet pinions, unequally as well as equally spaced. Differing from the conventional in that the sun and planet gears only are considered, the number of teeth in the ring gear is determined by

$$T_R = T_S + 2\,T_P$$

where the subscripts R, S and P refer to the ring, sun and planet gears respectively.

Ratio, k_1, of the revolutions of the sun gear to the revolutions of the spider is

$$k_1 = 2 + \frac{2\,T_P}{T_S}$$

The number of positions, n, in 180 degrees in which a planet gear can be placed is

$$n = T_S + T_P$$

Hence, for an equally spaced number of planets, n,

$$\frac{2(T_S + T_P)}{n} = \mathit{integer}$$

For an unequally spaced number of planets, again using n, select the nearest integer, A, to the above relation. Then the angle β between two planet gears must be

$$\frac{A}{T_S + T_P} \times 180^\circ = \beta$$

Thus in systems wherein planet pinions cannot be equally spaced, this method results in their location in as near equal spacing as possible. If approximation of equal spacing is not desired and planets are to be located at some other angle or angles, these may be calculated by assuming values of the integer A and the angle β which will result in $T_s + T_p$ being an integer. The proper tooth numbers for the sun and planet gears will then be determined by the desired ratio for the system.

In order to avoid possible interference between planets, the value of the ratio k_1 cannot exceed

$$\frac{2\,T_s-4}{T_s\left[1-\sin\left(\frac{180}{n}\right)\right]}$$

For planetary systems in which the planetary gears are mounted on fixed centers and the ring gear rotates (in the direction opposite to that of the sun gear), the ratio, k_2 of the revolutions of the sun gear to the revolutions of each planet is 1 less than given above, or

$$k_2 = 1 + \frac{2\,T_P}{T_s}$$

Maximum ratio, k_1

No. of Planet Gears	Ratio $T_s = 12$	$T_s = 15$	$T_s = 20$
2	∞	∞	∞
3	12.37	12.93	13.43
4	5.68	5.92	6.14
5	4.04	4.21	4.37
6	3.33	3.47	3.60

Ratio k_1 and Allowable Number of Planets*

No. of teeth in one Planet Gear	Number of Teeth in Sun Gear 12	13	14	15	16	17	18	19	20
12	4.000 2-3-4	3.846 2	3.714 2-4	3.600 2-3	3.500 2-4	3.411 2	3.333 2-3-4-5	3.263 2	3.200 2-4
13	4.167 2	4.000 2-4	3.857 2-3	3.733 2-4	3.625 2	3.529 2-3-4-5	3.444 2	3.368 2-4	3.300 2-3
14	4.333 2-4	4.154 2-3	4.000 2-4	3.867 2	3.750 2-3-4-5	3.647 2	3.556 2-4	3.474 2-3	3.400 2-4
15	4.500 2-3	4.308 2-4	4.143 2	4.000 2-3-4-5	3.875 2	3.765 2-4	3.667 2-3	3.579 2-4	3.500 2-5
16	4.667 2-4	4.462 2	4.286 2-3-4	4.133 2	4.000 2-4	3.882 2-3	3.778 2-4	3.684 2-5	3.600 2-3-4
17	4.833 2	4.615 2-3-4	4.429 2	4.267 2-4	4.125 2-3	4.000 2-4	3.889 2-5	3.789 2-3-4	3.700 2
18	5.000 2-3-4	4.769 2	4.571 2-4	4.400 2-3	4.250 2-4	4.118 2-5	4.000 2-3-4	3.895 2	3.800 2-4
19	5.167 2	4.923 2-4	4.714 2-3	4.533 2-4	4.375 2	4.235 2-3-4	4.111 2	4.000 2-4	3.900 2-3
20	5.333 2-4	5.077 2-3	4.857 2-4	4.667 2	4.500 2-3-4	4.353 2	4.222 2-4	4.105 2-3	4.000 2-4-5
21	5.500 2-3	5.231 2-4	5.000 2	4.800 2-3-4	4.625 2	4.471 2-4	4.333 2-3	4.211 2-4-5	4.100 2
22	5.667 2-4	5.385 2	5.143 2-3-4	4.933 2	4.750 2-4	4.588 2-3	4.444 2-4	4.316 2	4.200 2-3-4
23	5.833 2	5.538 2-3-4	5.286 2	5.067 2-4	4.875 2-3	4.706 2-4	4.556 2	4.421 2-3-4	4.300 2
24	6.000 2-3-4	5.692 2	5.429 2-4	5.200 2-3	5.000 2-4	4.823 2	4.667 2-3-4	4.526 2	4.400 2-4
25	6.167 2	5.846 2	5.571 2-3	5.333 2-4	5.125 2	4.941 2-3-4	4.778 2	4.632 2-4	4.500 2-3

* The upper figure of each pair is the ratio k_1; the lower figures indicate the number of planets which may be used equally spaced.


News
FORMER "DRY SPOT" Now Gets OIL A-PLENTY
Pins and Bushing Surfaces Fed By Exclusive Design of MORSE ROLLER CHAIN
Oil Gets In-Wear Stays Out!
That's NEWS! That's *good* news! For the pin and bushing surfaces of roller chain used to be the trouble spots, the place where failures frequently occurred because of insufficient lubrication.
But not with Morse Roller Chain now. Channels in the bushings lead oil through the sideplates down to the pin and bushing surfaces. The Morse design puts a "force-feed" lubrication system in every link. Oil gets in—wear stays out.
Get the advantages of Morse longer life, smoother operation, higher efficiencies for *your* roller chain drives. Morse roller chains are interchangeable with other roller chains, and they're available in all standard sizes, single and multiple widths. Call in the Morse man near you, or write direct to Morse, Ithaca.
SILENT CHAINS ROLLER CHAINS FLEXIBLE COUPLINGS CLUTCHES
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Professional Viewpoints

MACHINE DESIGN welcomes comments from readers on subjects of interest to designers. Payment will be made for letters and comments published

" . . . help save time"

To the Editor:

Your recent inauguration in your May issue of the department "Engineering Data Sheets" is really worth while.

I venture to say that practically every engineer has gathered together, over a period of years, data from many sources. These data have helped him to save time in some cases, while in others they have increased his store of ideas.

Without question your readers will welcome this new feature in an already well-edited engineering periodical.

—C. E. SCHIRMER, *Chief Engineer*
Robbins & Myers, Inc.

" . . . data for filing"

To the Editor:

Please send me data sheets of "Mathematical Solution of Four-Bar Linkages" as published in your July, 1941 edition. I would appreciate receiving these as soon as possible for my files.

—W. M. WATSON, *General Engineering Dept.*
The Heald Machine Company

A limited number of clip sheets of each of the data sheets in Mr. Talbourdet's series still is available. These will be furnished gladly upon request.—ED.

" . . . necessitates better design"

To the Editor:

As brought out by Messrs. Owen and Kirkish in their article entitled "Frequent Starting and Reversing Motors," it is important to keep down the WR^2 of the driven parts, particularly on fast-running shafts. It is equally important to remember that the increased acceleration and deceleration provided by modern reversing motors makes necessary better design, lubrication, and workmanship in the bearings and drive gears to which the motor is connected. Otherwise a very disagreeable "knock" in the drive will be in evidence.

—JOHN M. WALTER, *Chief Engineer*
The G. A. Gray Co.

" . . . for increased personnel"

To the Editor:

I have read with interest the article in your May issue by H. T. Pentecost, "Specifying Design Department Materiel." It was particularly interesting in that we were confronted recently with the problem of accommodating an increased personnel and it became necessary to utilize our floor space to more advantage. This problem was solved by designing special drafting board equipment. The accompanying photograph shows a board and some of its features.

The board itself is similar to that which has been described in your magazine. However, we feel that

several features of our unit are noteworthy. The board and drafting machine are counterbalanced so that the whole device may be moved up or down with ease. A small tray attached to the lower edge of the board accommodates personal instruments such as pencils, dividers and erasers. The top of the reference table is adjustable and provides writing space for the individual immediately ahead of the drafting board. There is also a sliding shelf which can be pulled out and used for writing material lists, etc. When drawing, this shelf is easily pushed out of the way.

Altogether approximately 50 of these vertical boards have been made and found to be successful. Three of the new boards occupy about the equivalent floor space which formerly was taken by two of the horizontal type. In this way we were able to accommodate successfully the increased personnel and, at the same time, to obtain an increase in efficiency.

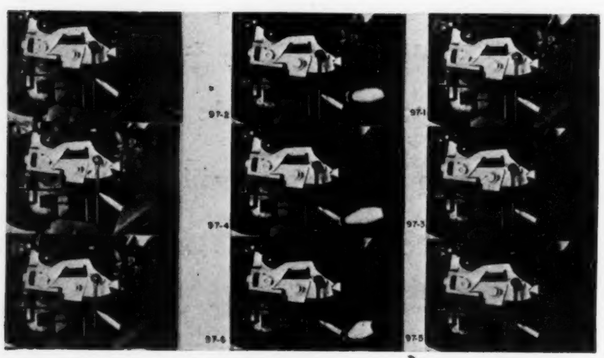
—W. W. Clark
Baker Perkins Inc.

" . . . for study of mechanisms"

To the Editor:

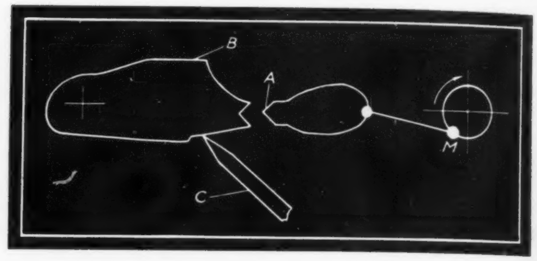
Referring to the article "Stopping Motion for Design Data" in your June issue, I would like to comment on another interesting application of the stroboscope for the study of rapid mechanisms.

In the article are given several examples of single-flash photography, as well as one showing how successive exposures were made on an ordinary movie camera. The method we have used is somewhat similar. We focus a camera upon the mechanism under study and then flash the light at successive intervals. Each flash produces a picture



showing the progressive position of the mechanism on separate pictures. Then these pictures are pasted in one continuous strip, thereby resulting in a series in which each succeeding picture shows the action of the mechanism. The series can be studied at leisure and the flaws in the mechanism's operations observed easily. A good example of this type of stroboscopic analysis is given on the accompanying section of a study sheet. In this series are shown the theoretical and correctly adjusted operations of a bobbin transfer mechanism as used in transferring bobbins in a modern loom. The operation of this mechanism is explained in a crude way by means of the sketch.

The latch stud *C* moves lever *B* upward and, as the crank carrying pin *M* revolves, it causes the dog *A* to engage the notch in the lever. The dog and lever are then moved backward together until pin



M has reached its farthest position. After this, as point *M* moves away from its dead center position, the dog and lever should move back together until the lever has reached its limits of travel and the dog and lever should then separate, and the lever be lowered by stud *C*.

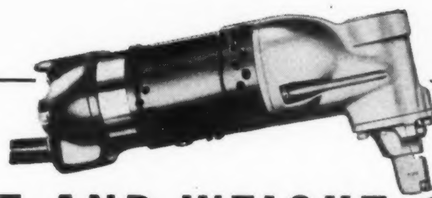
The theoretical operation of this mechanism is shown at the left in the photograph. At the right the mechanism is set to operate using a recommended setting of 1/32-inch clearance between the transfer hammer and the ingoing bobbin. (The hammer is not shown in sketch, but is so pivoted and attached to lever *B* that the backward motion of this lever moves the hammer down, thus causing a new bobbin to be inserted into the shuttle and the old one to be pushed out). It is seen that, regardless of whether a bobbin is actually being transferred or not, the stud never interferes with the lever.

However, if the mechanism is not properly adjusted and the clearance between the hammer and the bobbin is increased to ¼-inch, then the story is different. If the hammer pushing the new bobbin into the shuttle has to expel a deep-seated bobbin already in the shuttle in order to insert the full one, then the hammer does not overrun. That is, the resistance offered by the bobbin in the shuttle to the hammer pressing in a new shuttle is sufficient to prevent the hammer's inertia from the "snapping" of the lever by the stud. If on the other hand, there is no bobbin in the shuttle, then the inertia of the hammer allows the lever to snap by the stud and the lever "hangs up." This action was also studied photographically.

Smaller figures in the lower left-hand corner of each "shot" are the reference negative numbers, horizontal views are corresponding angular position of the crank at the time the picture was taken. This crank angle applies to all of the "shots" in the same horizontal line.

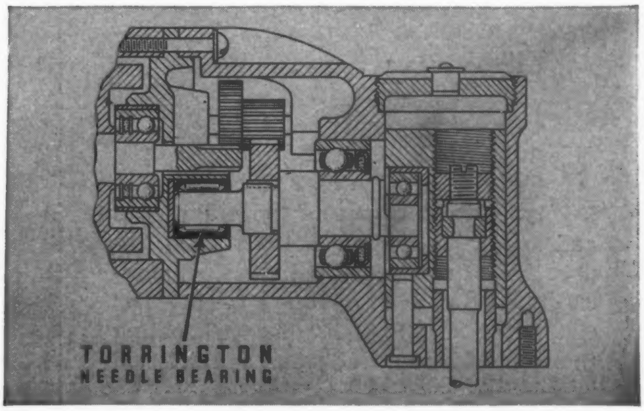
—V. F. Sepavich
Crompton & Knowles Loom Works

Correction—In the article "Designing for Hydraulic Variable Speed," beginning on page 41 of the July issue, *Fig.* 1 should be *Fig.* 2 and vice versa in order to agree with the manuscript.



Thor CUTS SIZE AND WEIGHT OF POWER TOOLS WITH ANTI-FRICTION NEEDLE BEARINGS

THIS COMPACT THOR hand-model U1N Nibbler packs in plenty of power! It weighs only 3¾ lbs., yet does the work of a portable power shear and, in addition, efficiently cuts irregular shapes, follows lines and contours, and even cuts inside shapes on a radius as small as 1½ in., starting at a drill hole.



HOW IS IT MADE JUST A HANDFUL? Here's what Mr. G. Larson, Chief Designer of Independent Pneumatic Tool Co., says: "The Torrington Needle Bearing's very small O.D. makes possible our compact gear case. And it gives us good anti-friction service without trouble, in the Nibbler and many other tools."

THE NEEDLE BEARING (indicated by Mr. Larson's pencil) is fairly inaccessible when the tool is assembled, but no extra lubrication system is necessary because a large supply of grease is retained and evenly distributed in the reservoir formed by the close-fitting lips of the bearing's race.

OPERATING AT HIGH SPEEDS the Torrington Needle Bearing shows its ruggedness on jobs like this. Its precision-ground rollers and hardened outer race form a self-contained unit of high radial load capacity that is easily, inexpensively installed in almost any type housing. Initial cost is also surprisingly low.

If you have a bearing problem where high load capacity, small size, light weight, ease of assembly and lubrication are vital considerations, and low cost is an important factor, investigate the Torrington Needle Bearing. Our Engineering Department will

be glad to work with you in adapting its advantages to your product. For details, write for Catalog No. 109. For Needle Bearings to be used in heavier service, write our associate, Bantam Bearings Corporation, South Bend, Indiana, for Booklet 103X.

THE TORRINGTON COMPANY, TORRINGTON, CONN., U. S. A. • ESTABLISHED 1866

Makers of Needle and Ball Bearings

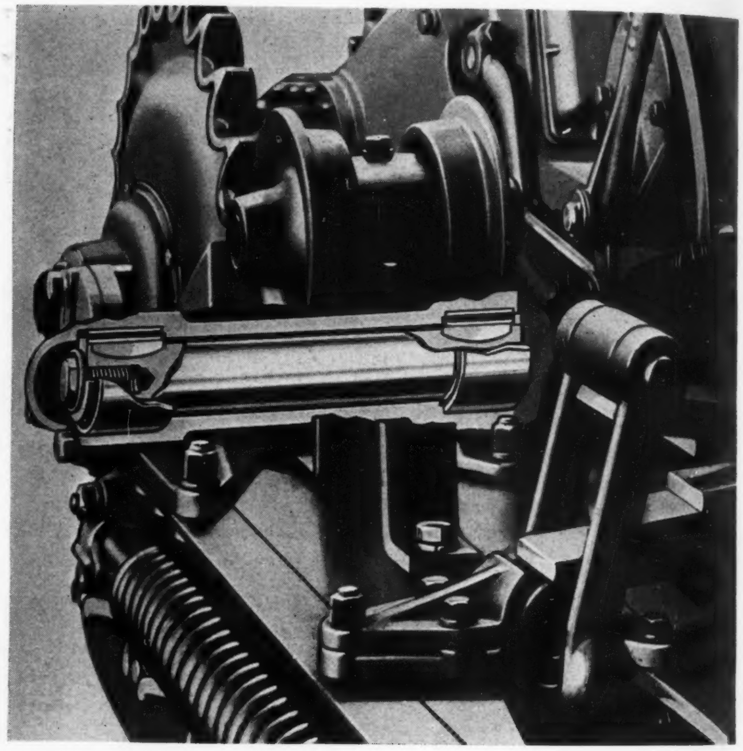
New York Boston Philadelphia Detroit Cleveland Chicago London, England

1866 75 YEARS OF MANUFACTURING PRECISION PRODUCTS 1941

TORRINGTON NEEDLE BEARING

Applications

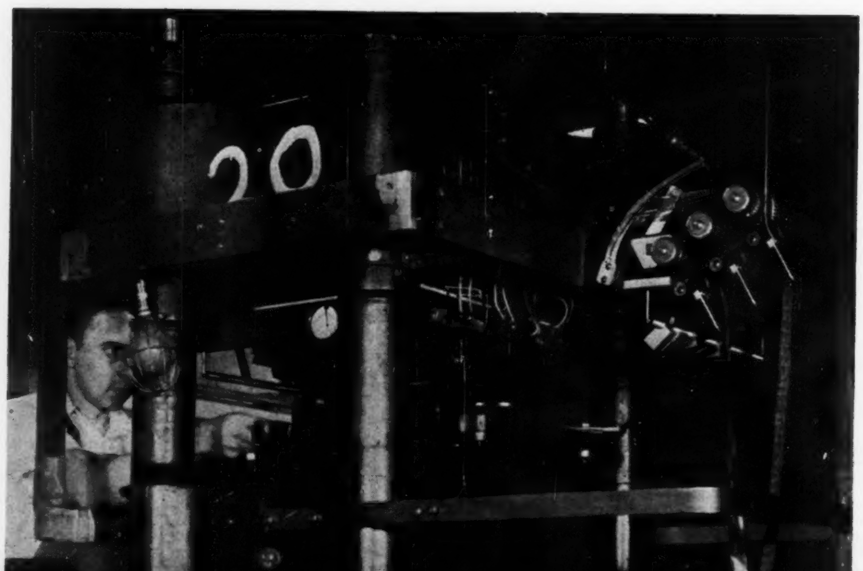
of Engineering Parts and Materials



Eliminates Excess Wear

FASTENED to the transmission, the pivot axle of Allis-Chalmers track-type tractors carries the entire load and oscillating motion of the trucks. Application of rubber-backed bearings manufactured by Harris Products Co. has eliminated all necessity for lubricating this axle. In addition, wear and friction is reduced.

Two bearings are used for each truck—one at each end of the pivot bearing assembly. Oscillating motion of the trucks is absorbed by the rubber, eliminating friction which otherwise would be present if relatively moving, co-acting metal bearing surfaces were used. Also, metal bearings at this point would introduce the necessity of providing dirt and grit seals to prevent abrading and seizing of pivot points.

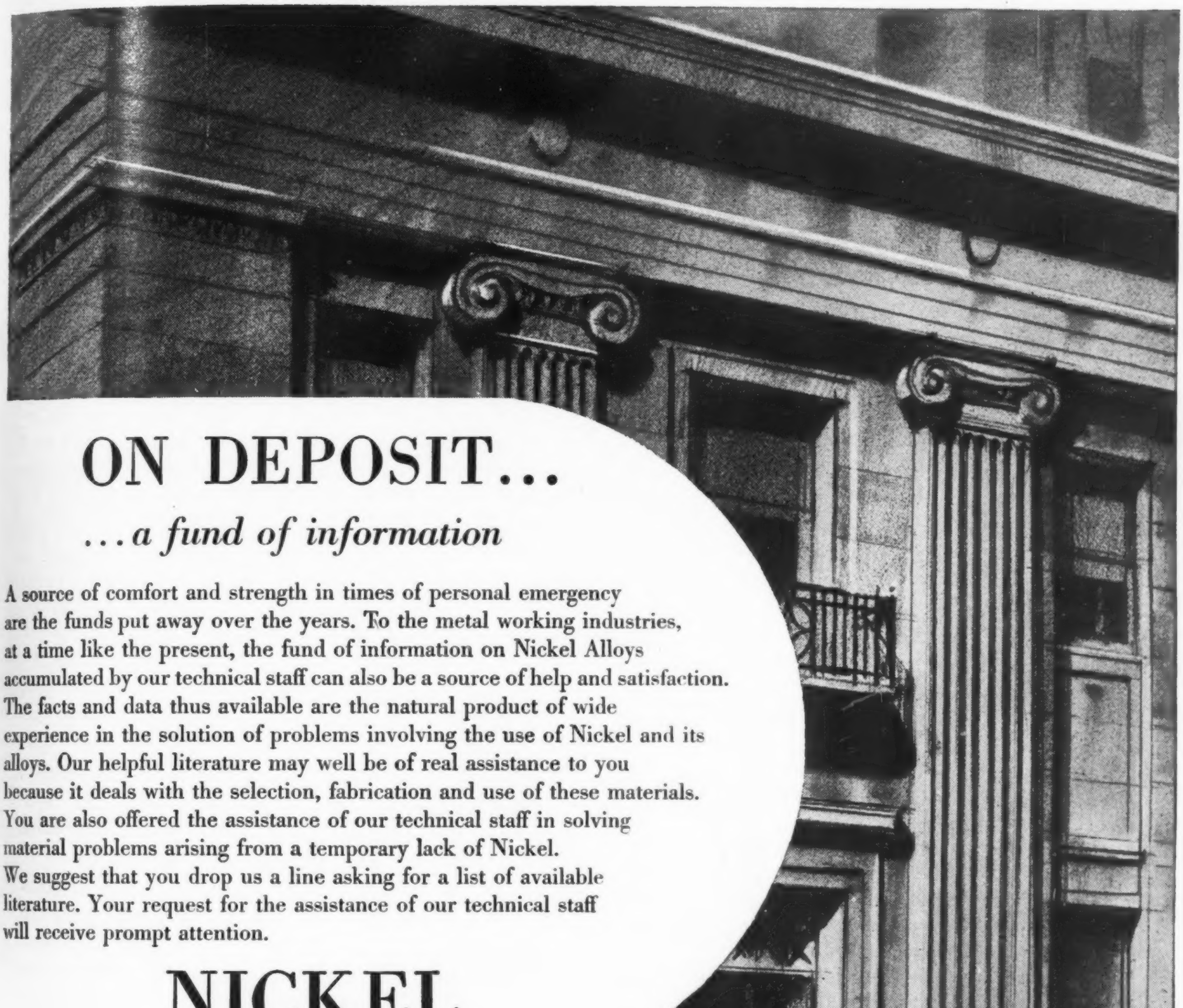


Embosser Uses Strip Heaters

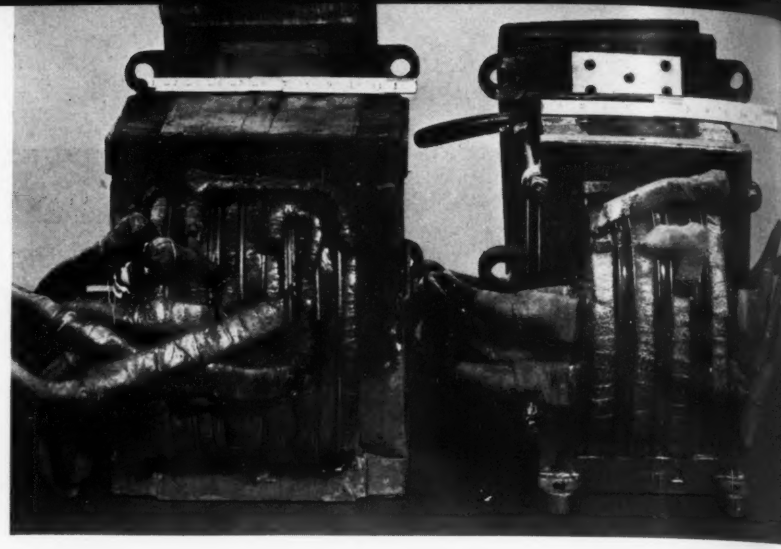
TEMPERATURES of from 300 to 400 degrees Fahr. are required in stamping machines for embossing and applying gold leaf. Previous method, employing gas chambers located above the top press plate, resulted in difficult temperature regulation with risk of firing the gold leaf and considerable operator discomfort.

Electric strip heaters were installed in long, thin, rectangular tubes inserted into the gas chambers. The area between the tube wall and the walls of the chamber was plugged with babbitt.

As a result of this installation of G. E. heaters, the Wilson Jones Co. was able to speed up the heating considerably since each of the units is individually controlled. Not only are


ON DEPOSIT...
... a fund of information
A source of comfort and strength in times of personal emergency are the funds put away over the years. To the metal working industries, at a time like the present, the fund of information on Nickel Alloys accumulated by our technical staff can also be a source of help and satisfaction. The facts and data thus available are the natural product of wide experience in the solution of problems involving the use of Nickel and its alloys. Our helpful literature may well be of real assistance to you because it deals with the selection, fabrication and use of these materials. You are also offered the assistance of our technical staff in solving material problems arising from a temporary lack of Nickel. We suggest that you drop us a line asking for a list of available literature. Your request for the assistance of our technical staff will receive prompt attention.
NICKEL
THE INTERNATIONAL NICKEL COMPANY, INC.
67 WALL STREET NEW YORK, N. Y.

correct temperatures easily maintained but gas fumes and a serious fire hazard has been eliminated. Since the operator works directly beside the heaters, conditions are much improved.

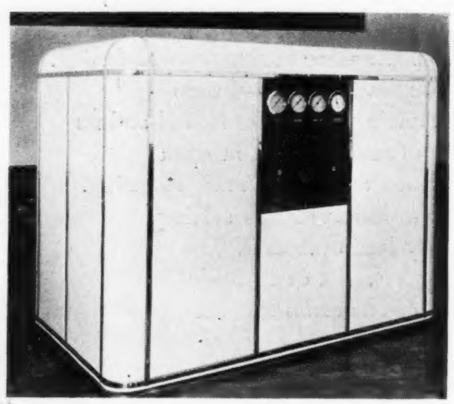


Glass Effects Size Reduction

IN addition to already realized reductions in size of electric motors brought about by use of Fiberglas insulation, this same material when applied to other electrical equipment accomplishes similar desirable results. Illustrated is a "before and after" picture of a transformer. Superimposed scales indicate the degree of size reduction that has been attained.

In addition to withstanding high temperatures, humidity and corrosive vapors, reserve strength is provided which is capable of taking ordinarily excessive overloads without the risk of failure.

Currently important also is the fact that this insulating material is made from resources which are plentifully available within our own borders so that supply is completely independent of foreign sources. A further advantage not to be ignored is the ease with which it can be applied as electrical insulation. One manufacturer estimates that a time saving of almost 40 per cent was achieved over a similar application of asbestos.

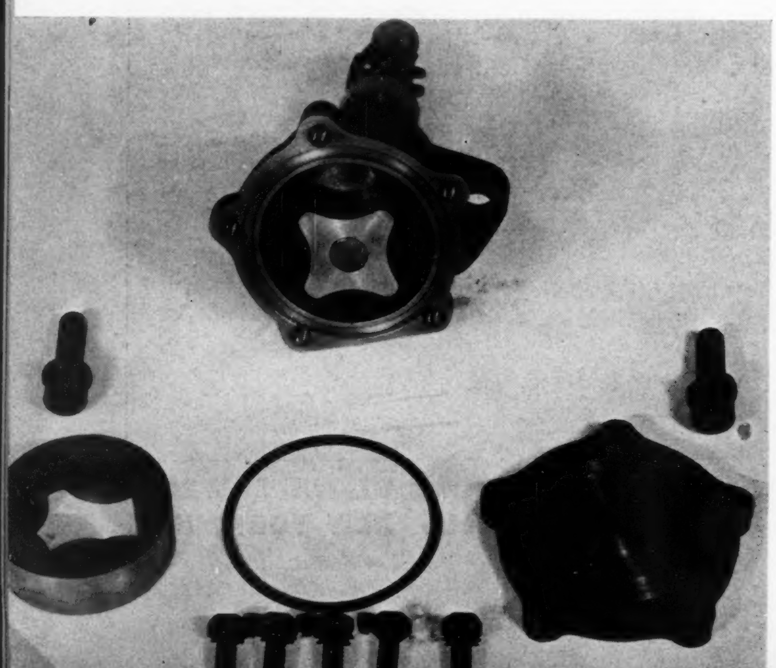


Built by New System

CAPABLE of being assembled without rivets or welds, a new prefabricated method of construction has been applied to the Chil-Derator beverage cooler illustrated. Called Lindsay Structure, the system permits of rounded corners and a smooth exterior. The beverage cooler is enameled and has chrome trim.

Under present conditions it may be all but impossible to obtain the necessary dies to build a similar structure of sheet metal. In addition the fastening methods conventionally used for sheet metal assemblies may, under high production schedules, cause another bottleneck. For these reasons this prefabricated, die-formed method of construction is finding extensive application.

In accord with its progressive exterior construction the cooler illustrated incorporates also a novel system of refrigeration. The dual functions of water cooling and deaeration are performed. This is accomplished by passing steam over the water in the cooler, thereby causing a fraction of it to evaporate, thus cooling the remainder. The water so cooled and freed of dissolved and absorbed gases is used as the refrigerating medium for bottled beverages.



Pump Has Sintered Gears

BOTH inner and outer rotors of gerotor-type pump are made of sintered powder metal. Each is die-shaped under high pressure, heat treated, sized on the cylindrical surfaces by burnishing dies and coined on the faces to exact dimensions. By these processes the extremely critical contours of the rotor tooth profiles are accurately made, requiring no machining of these Chrysler pump parts.

Outer rotor lobes are five circular arcs equally spaced. In the designing, the inner rotor is generated from the outer by revolving it eccentrically to the outer and, at the same time, turning it 1/5 of a revolution for each revolution of the inner rotor axis about the outer rotor axis. This generates the four-toothed inner rotor. From the design thus obtained the dies for molding are made.

Same Size – Larger or Smaller
Partial or Complete Duplication

Masterrepro is a sensitized water-proof tracing cloth. Using an original tracing, you will get a fresh duplicate equal but more durable than the hand made original drawing. You can erase it, you can add lines or numerals in India ink or pencil. Think of the possibilities in your own drafting when Masterrepro takes the place of hand redrawing for any of the following uses . . .

1. Identical duplicates.
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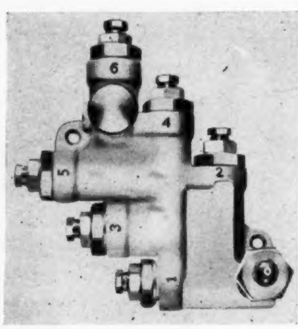
Reproduction Print Service, Inc.
Oklahoma City—The A & E Equipment Co.
Omaha—Standard Blue Print Co.
Philadelphia—Philadelphia Blue Print Co.
Pittsburgh—American Blue Printing Co.
Portland—J. K. Gill Co.
St. Louis—Service Blue Print & Photo Copy Co.
Salt Lake City—Salt Lake Blue Print & Supply Co.
Seattle—Kuker-Ranken, Inc.
Tampa—Office Equipment Company
Toledo—Toledo Blue Print & Paper Co.
Tulsa—Triangle Blue Print & Supply Co.
Washington, D. C.—R. E. MacMichael
Wichita—City Blue Print Co.

New PARTS AND MATERIALS

Progressive Lubricating System

CENTRALIZED to permit high pressure grease lubrication to any number of bearings on industrial equipment, a new progressive lubricating system has been announced by Alemite division, Stewart-Warner Corp., 1812 Diversey Parkway, Chicago. From a central point the system delivers a predetermined quantity of lubricant to from three to twenty bearings. Where a larger number of bearings need lubrication, this system may be employed in relays, each system or relay lubricating a maximum of twenty bearings. Any type of manually or



power-operated high pressure grease gun may be used. The system, which can be easily installed, offers four major features: All bearings in one piece of equipment can be lubricated simultaneously from one central point; inaccessible bearings are lubricated; lubrication of complete machinery is speeded; and the machines may be lubricated while operating.

Circuit Interrupter Introduced

FOR use in locations exposed to dust or water spray, or where water or dust-tight nonfusible switches were formerly used, a circuit interrupter has been introduced by Westinghouse Electric & Mfg. Co., East Pittsburgh. NEMA Types 4 and 5 enclosures are furnished in either cast aluminum or cast iron, both with an aluminized finish. A closely machined fit between handle shaft and bushing with heavy rubber gaskets between enclosure and cover, make them watertight. With current ratings ranging from 50 to 600 amperes, the interrupter can be furnished in two or 3 poles at 250 to 600 volts alternating current and 125 to 250 volts direct current. To interrupt a circuit adjusted for 5000 to 10,000 amperes, depending on frame size, a high interrupting capacity is provided. De-ion arc quenchers confine, divide and extinguish arcs al-

most instantly. "Quick make—quick break" action is obtained by a toggle mechanism operating the contacts.

Switch Has Many Mountings

WITH a variety of mounting positions as its most useful feature, a new splashproof, malleable housed, precision switch is announced by Micro Switch Corp., Freeport, Ill. This switch, which can be conveniently mounted from any one of four sides, can be used as an interlock, limit or pushbutton switch where closely held operating points and movement differentials are needed and where any

splashing may occur. For mounting there are two tapped holes on each of four sides, making pos-

(*Continued on Page* 94)

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Submit YOUR bearing problems to us, for study and engineering recommendations—without obligation. Write for the Catalog.

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PRECISION BALL • ROLLER AND THRUST BEARINGS

(*Continued from Page 88*)

sible the mounting of the switch directly on a machine frame in practically any position—whether the switch has a roller arm, cross-roller arm, or push-rod plunger type of actuation. If lug or foot mounting is desired, 3/16-inch thick steel mounting plates can be furnished. Switching element is a bakelite micro switch providing precision operation, ample overtravel and long life, and is listed by Underwriters' at 1200 watts up to 600 volts alternating current. The aluminum roller arm is adjustable through 360 degrees with the roller riding on an oilless-bronze bearing. Overtravel on this switch is 90 degrees. Cover is removable and the switch is so mounted inside the housing that it may be removed easily for inspection or change.

Synthetic Enamels for Castings

NEW and fast-drying synthetic enamels for machine tool castings, announced by Sherwin-Williams Co., Cleveland, are said to cut many hours and even days from the finishing time formerly required on large tools, and also give greater resist-

ance to cutting compounds. Four coats of this new enamel can be applied quickly enough to permit work to be shipped the same day it is finished and assembled. A typical drying schedule follows: First, the spray or brush coat zinc chromate primer dries in 15 to 30 minutes; second, the machine filler, in 4 to 5 hours; third, the sealer gray, in 15 to 30 minutes; and fourth, the machine tool gray, in 15 to 30 minutes, and drying and crating, 1 to 2 hours, or a total of 5¾ to 8½ hours.

Offer Photoelectric Control

PHOTOELECTRIC inspection and registration control, Type A80, introduced by Photoswitch Inc., Cambridge, Mass., offers a simplified system for accurately controlling or inspecting cutting and printing operations on cellophane, paper, cloth, tin, metalfoil, etc. It can be used also for detecting presence or absence of labels on cans and proper location of labels on goods and similar applications. Extreme sensitivity with high-speed operation is

Save Space by using FORMEX

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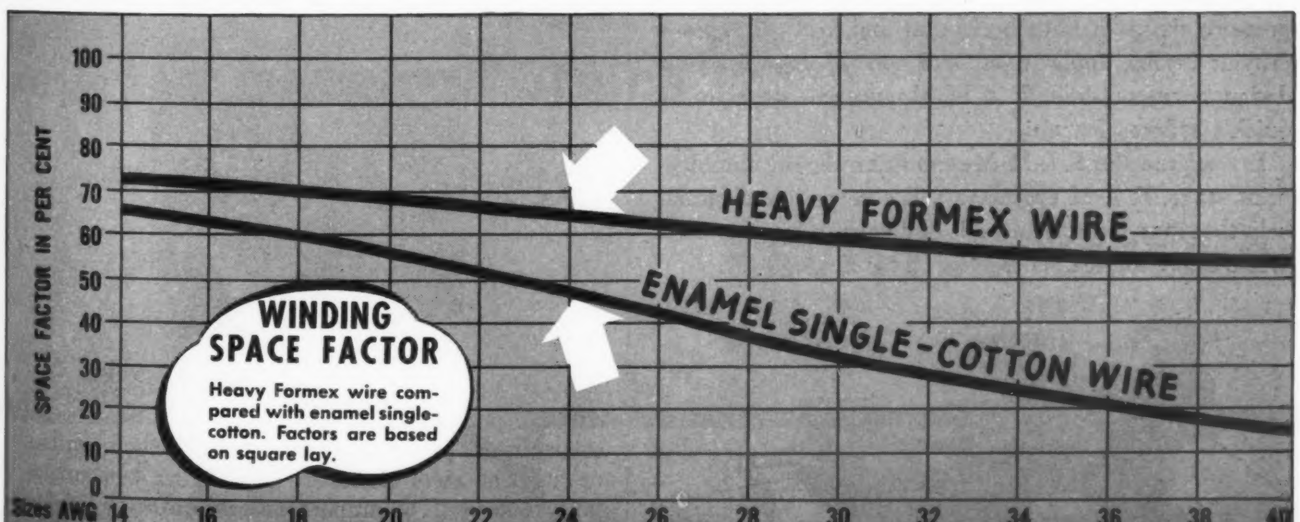
MAGNET WIRE

ONE of the greatest advantages in the use of Formex magnet wire is its high winding-space factor, which means space saved. For example, compare Formex wire with enamel single-cotton. In size No. 24 Awg (see curves), Formex wire has a winding-space factor of 64 per cent compared with 47 per cent for enamel single-cotton—36 per cent higher!

This suggests what can be accomplished with Formex magnet wire: First, by redesigning a machine you can have a smaller, more compact unit where size is important; second, by keeping the same size unit you can increase all-round performance through greater slot density, that is, a larger number of turns for a given space.

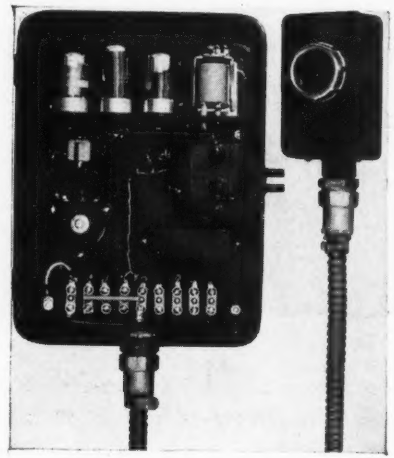
Consider, too, other distinct advantages in the use of Formex magnet wire: You get *more feet per pound;* it *resists many solvents;* and it has *high dielectric strength* and *great abrasion resistance.*

For further information about Formex magnet wire, or for any other data on G-E insulated wire and cable, get in touch with the G-E office near you or write direct to General Electric Company, Schenectady, N. Y.



GENERAL ELECTRIC

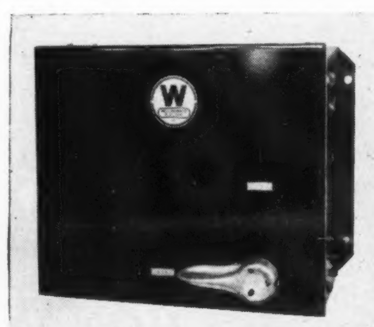
combined in the control, which observes registration marks from penetration of light through transparent and translucent materials as well as from reflection from opaque materials. The control will



detect an impulse of as short a duration as .001-second. Upon receiving an impulse the control relay operates and remains in operation until the controlled circuit has completed its function, at which time the unit is reset automatically. All amplifying tubes are standard vacuum type.

Semimagnetic Motor Starter

TO FURNISH low cost means of starting and stopping, a new direct current semimagnetic motor starter has been designed especially for the mining field by Westinghouse Electric & Mfg. Co., East Pittsburgh. These 1 to 15 horsepower starters provide overload and circuit protection on 230-550 volt direct current lines, and are used on motors,

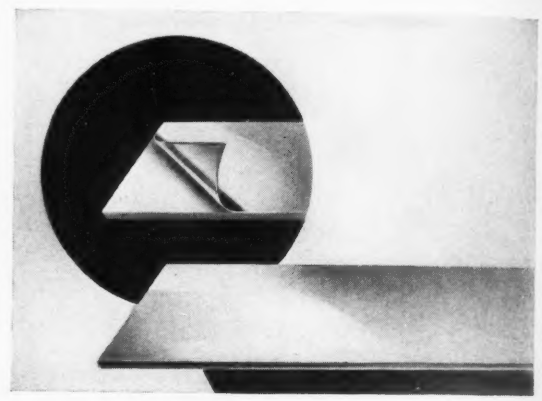


driving pumps and compressors. Automatic acceleration is provided in this starter which consists of a sheet steel enclosure containing one single-pole, 600-volt, 60-ampere, front operated De-ion line switch, a small ebony asbestos panel, an accelerating contactor, and a resistor. When switch is closed motor is started with resistor in series with armature to limit starting current. The accelerating contactor coil is connected across the motor armature. As the motor accelerates, the voltage across the armature increases and causes the accelerating

contactor to close at the proper time, short circuiting the resistor and connecting the motor across the line.

Laminated Shim Stock Is Larger

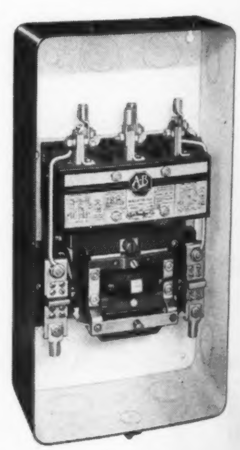
LARGER one piece shims can now be cut from the new 7-inch laminated shim stock now in production by laminated Shim Co. Inc., Glenbrook, Conn. The stock can be furnished in 7 x 36-inch dimensions in addition to the 6 x 36-inch sheets.



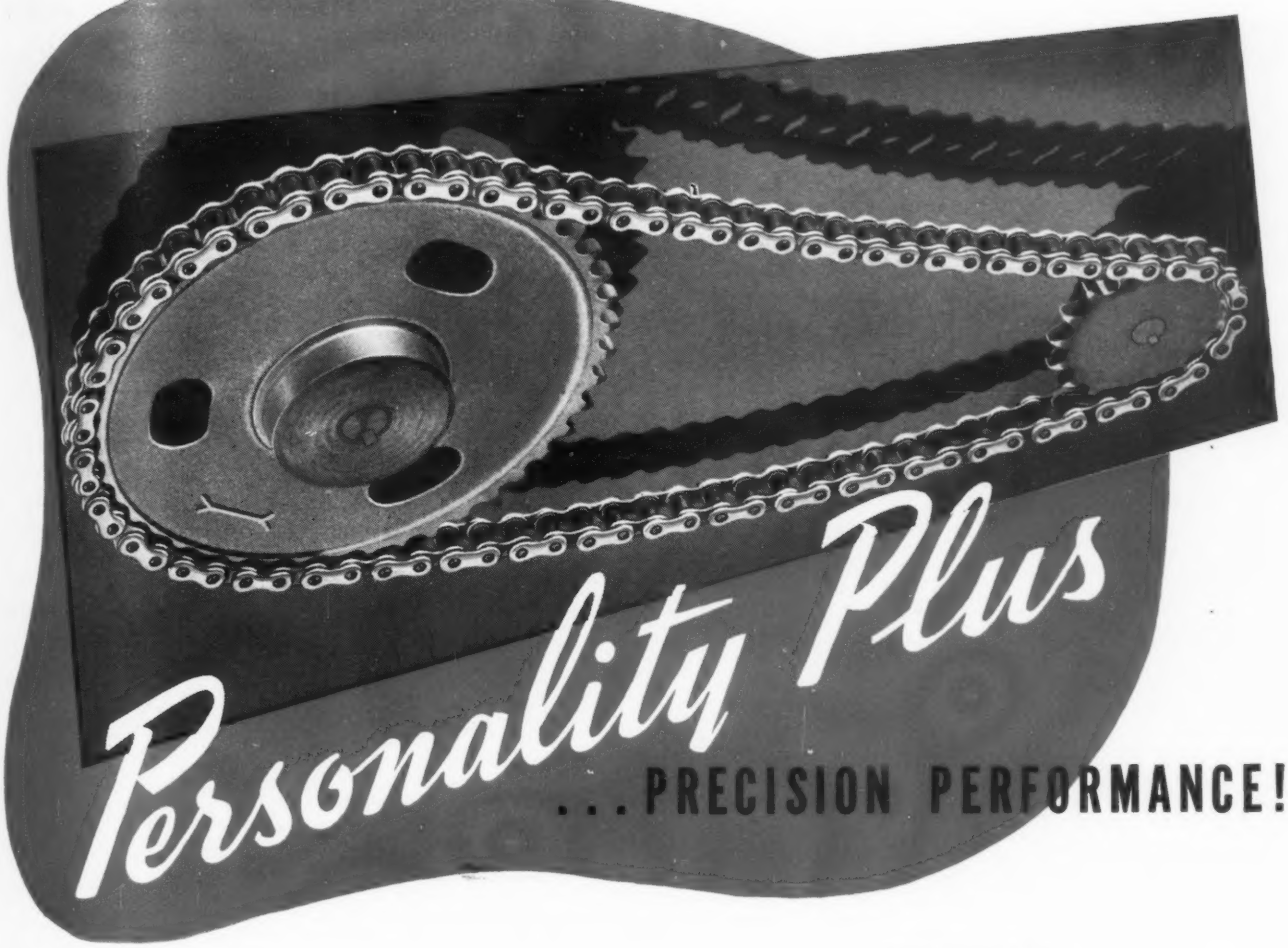
The new width permits the manufacture of larger jointless shims by the company as well as affording the same facility to users who cut their own shims. Overall thicknesses of the new sheets are from .006 to .125-inch. These may be obtained in all-laminated sheets with a choice of .002 or .003-inch thick laminations; or various thicknesses may be had partly laminated and partly solid.

Solenoid Starter Developed

LARGER starters with solenoid switch structures are being offered by Allen-Bradley Co., 1311 South First street, Milwaukee, to replace its clapper starter. The new unit has a maximum horsepower rating of 50 horsepower, 220 volts and 100 horsepower, 440-550-600 volts. This new, size 4, solenoid starter incorporates all the features of the company's smaller solenoid starters, and no contact cleaning, filing or dressing is necessary because of cadmium silver contacts. There are no bearings, pivots or hinges. The starters can be closely grouped without danger of flash-over between



switches as the double break cadmium silver contacts are totally encased in an arc hood, each pole of the switch having its individual arc cham-



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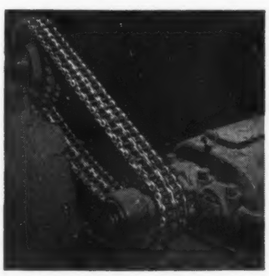

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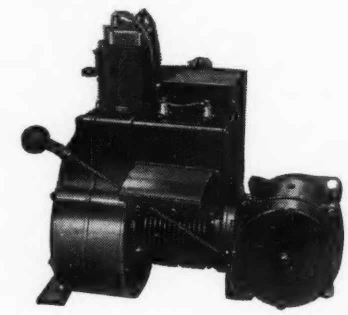
LINK-BELT Silverlink ROLLER CHAIN

ber. By employing solenoid design, the size of the starter has been greatly reduced and, at the same time a generous amount of wiring space is provided. Arc-interrupting capacity is high; currents of at least ten times the maximum horsepower rating are easily interrupted. Starter is mounted on a self-insulated metal base-plate which may be mounted on any metal surface without extra insulation. It can be furnished with or without an enclosure.

Portable Industrial Engine

TWO-CYCLE, portable, aircooled industrial engines for chain saws, pumps, generator sets, fire-fighting equipment, compressors, tractors, mowers, motor boats, etc., have been introduced by Kiekhaefer Corp., Cedarsburg, Wis. This engine can also be used for any portable equipment for the Army and Navy, lumber, mining, pulpwood and



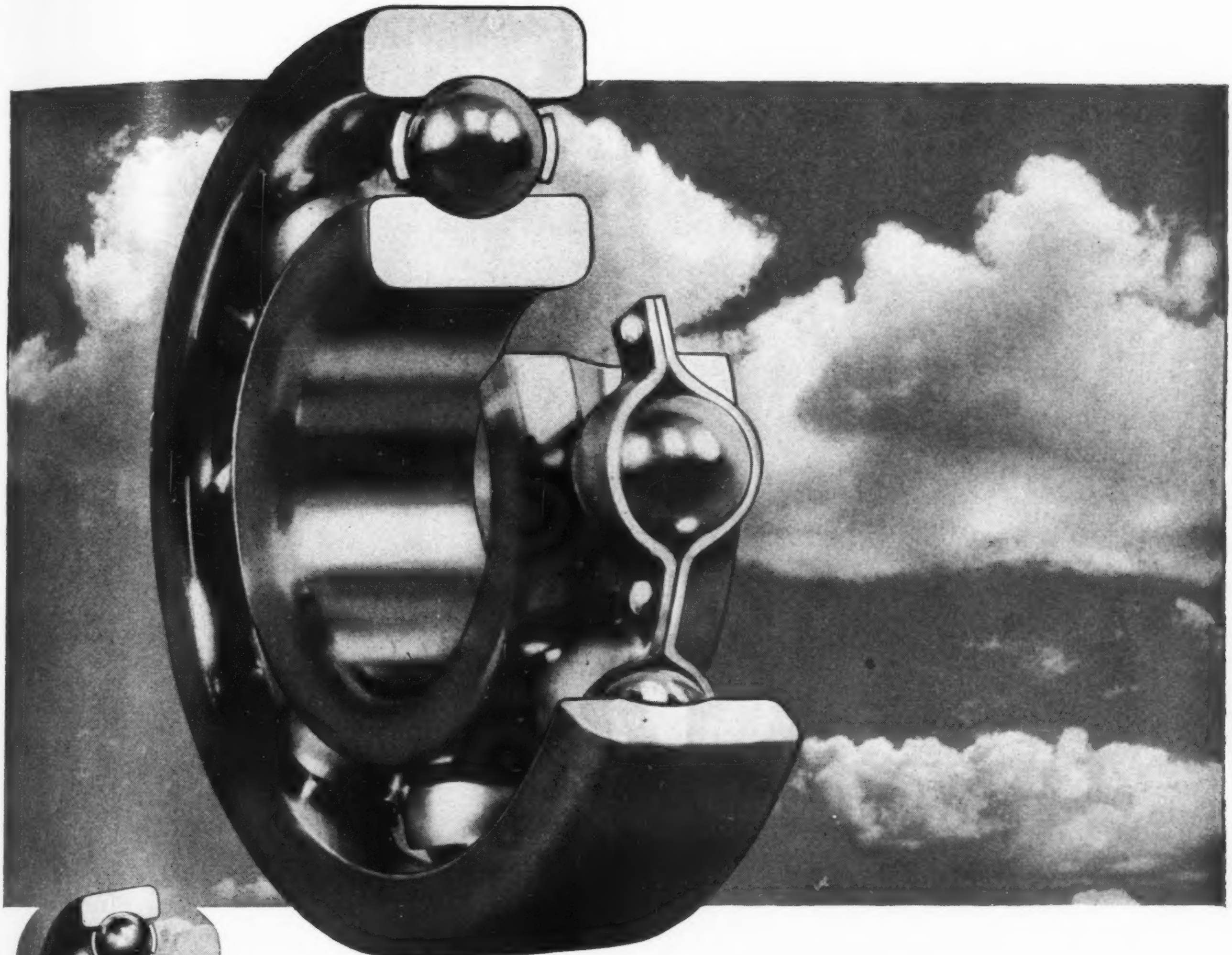
other industries. On recent tests the engine demonstrated easy starting, freedom from carbon and spark plug trouble, and cool running after continuous use. The line consists of a 1-cylinder, 2½-horsepower engine, a 2-cylinder, 5-horsepower engine, and a 4-cylinder model which develops 10 horsepower. Motors are light in weight; the 5-horsepower model with clutch and transmission weighs only 70 pounds. Capable of being mounted in either vertical or horizontal position, the engine may be obtained complete with clutch, in various gear ratios, and with several transmissions.

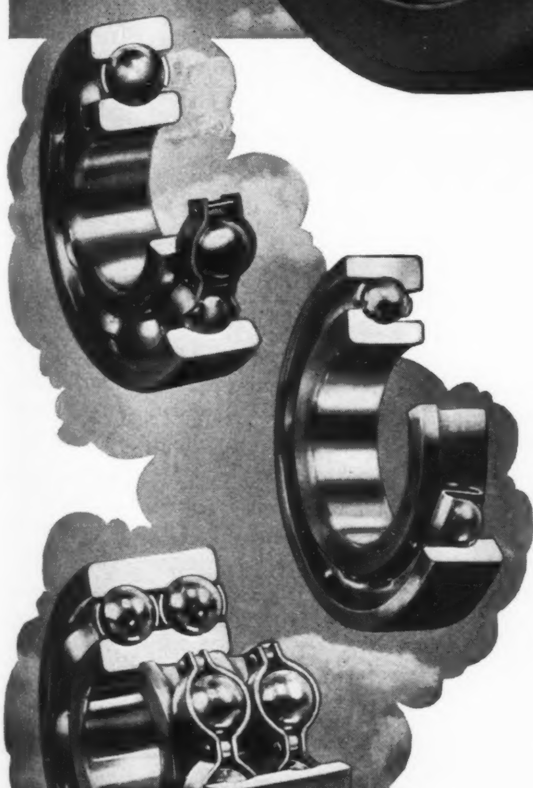
Varnish-Saturated Tubing

VARNISH-SATURATED tubing for insulating wires or increasing dielectric strength of insulated wires when used as leads, connecting wires, etc., is now available through the General Electric appliance and merchandise department Bridgeport, Conn. The tubing is made of closely woven cotton yarns impregnated with varnishes, and is available in three grades, in sizes 20 to minus 1½, either black or yellow.

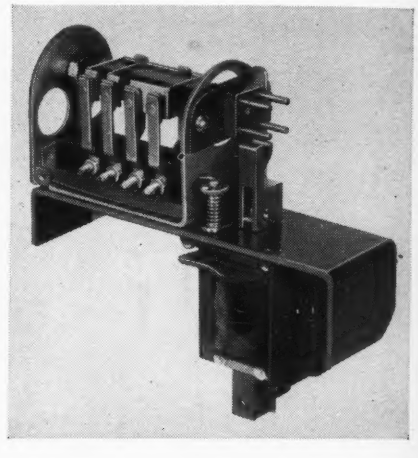
Switch Has Multiple Contacts

SOLENOID-OPERATED, multiple-contact switch, originally developed for motor reversing service without use of relays, by Vallen Inc., Akron,





O., is capable of application to "on" and "off" circuit control up to four contacts. It can be used in many applications in controlling 110 or 220-volt single or polyphase electrical circuits with or without relays. Unit will control up to 15 amperes at 220 volts or 25 amperes at 110 volts alternating current. Solenoid is energized by momentary contact switch causing plunger to lift. Independent circuit energizing the solenoid may be 22, 110 or 220 volts. Momentary contact demand of solenoid



is .75 amperes at 220 volts. Toggle engages pin on rotor, turning it 90 degrees and making or breaking contacts. Wiping contact of plated copper contacts assures good electrical connection. Break is fast, reducing arcing to minimum.

Engineering Dept. Equipment

"Hairline" Accuracy in Machine

BUILT to "hairline" accuracy, the new drafting machine introduced by The Frederick Post Co., Hamlin and Avondale avenues, Chicago, embodies a protractor, vernier, T-square, scales and triangles, is easily operated, and obviates smudging and blurring caused by moving instruments over the drawing. A "Quick Flick" control, which by a touch of the thumb releases the scales for 15-degree automatic stops, or for intermediate stops and locking, provides simple positive control by the left thumb (right thumb on left-hand machines). Again, by a flick of the thumb, the scale can be set at any angle with the protractor and vernier reading of zero to zero. Simple in design, the hardened steel construction of the machine gives greater accuracy and wear. The index plate subject to greatest wear is notched to take up any wear automatically.

Two Drafting Tables Offered

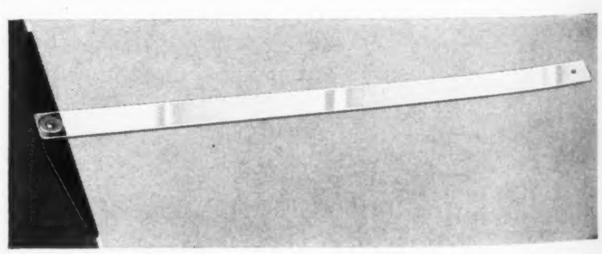
TWO new drafting tables—the Primo Metapost and the Metapost—have been announced by The Frederick Post Co., P. O. Box 803, Chicago. The Primo Metapost can be adjusted quickly and

smoothly by turning an operating handwheel to raise the working surface from 35½ inches to 43 inches. Table top may be tilted from front to back at an angle of 60 degrees by adjusting clamps. The Metapost has the same adjustment features with the exception of the handwheel control. To raise the top of this table requires loosening two thumb screws on the uprights supporting the table top,

permitting the working surface to be raised from 35½ inches to 43 inches. The top of this table may also be tilted. A reference shelf may be attached to both tables, which are available in eleven top sizes ranging from 31 by 42 inches to 48 by 96 inches.

T-Square Has Transparent Blade

RECENTLY developed and announced by Engineering Sales Co., Sheboygan, Wis., the new T-square is particularly adaptable where it is desirable to view the entire drawing without moving the T-square. Outstanding features include the use of plastics and a removable head construction. The head of the T-square is made of black laminated bakelite composition, 5/16-inch thick; blade is of



escolite, .09-inch thick. Both materials are made especially for this purpose, and will not warp, bow, chip, crack or split. Nor will the finish wear off, chip or peel. Eight cone-shaped pins are set into the head which engage in corresponding openings in the blade. No paper rabbet is necessary because head is held 1/16-inch away from the blade. A special die casting and single screw holds the entire assembly together. This provides for instant removal of blade for use as a straightedge. The T-squares are available in seven sizes.

MEN Of Machines

WELL known as a test pilot and chief engineer of the Glenn L. Martin Co., W. K. Ebel has been named vice president in charge of engineering. He was hitherto responsible for the engineering, as well as flight testing, of all bombers for the Army, Navy and the British. Approximately 1400 engineers of various classifications will serve under him. He will retain his title of chief test pilot.

A graduate of the Case School of Applied Science, he joined the Martin company in 1922 as draftsman and worked his way to the top of the engineering division. As assistant chief engineer of the company he was closely connected with the development of the China Clipper and her sister ships, and was also the first man to fly the China Clipper. Designs and developments of current bombers took place under Mr. Ebel's direction.

VICE president and chief engineer of the Detroit Edison Co., James Wentworth Parker has been nominated for president of the American Society of Mechanical Engineers. Mr. Parker has been a member of the Society since 1913 and has served on numerous committees.

Born in Auburn, N. Y., in 1886, he was prepared for college in the Louisville high

(*Continued on Page* 112)

(*Continued from Page* 106)

school, and graduated from Cornell university in 1908 with a degree of mechanical engineer. In 1935 he was awarded an honorary degree of master of science in mechanical engineering by the Detroit Institute of Technology. Mr. Parker served an apprenticeship after graduating from Cornell, first with De Kalb Power & Light Co. and then with Vincennes Street Railway Co. In 1910 he became boiler-room engineer with the Detroit Edison Co., and has been employed continuously by this company to the present time, with the exception of a year's leave of absence for war service. He has organized and directed various engineering projects of the Edison company, notably the design and construction of several of its electric power plants. In addition he is generally responsible for the operation of the company's electrical system and of its generating plants, central-heating plants and system. During the war Mr. Parker served as consulting mechanical engineer and head of the Nitrate division of the Ordnance department, United States Army.

♦

G. E. F. LUNDELL, who has served as vice president of the American Society for Testing Materials from 1939 to 1941, has been elected president of the society. Mr. Lundell is chief of the Chemistry division, National Bureau of Standards, Washington.

♦

W. E. GRIFFITHS will manage a new department, known as the Development Engineering department, for Allegheny Ludlum Corp., Pittsburgh. This department has been set up to assure full attention to the development of new processes and products for the future despite present demands on production. W. F. DETWILER JR. will act as assistant manager.

♦

W. C. SEALEY is the new engineer-in-charge of transformer design, transformer division, Allis-Chalmers Mfg. Co.

♦

DAVID C. PRINCE, manager of the commercial engineering department, General Electric Co., Schenectady, N. Y., was recently elected president of the American Institute of Electrical Engineers.

♦

CARL A. WOERWAG, who has been with the Link-Belt Co. since 1910, beginning in the drafting room, replaces LAURENCE M. EWELL as export manager at the company's New York office.

♦

R. L. MCILVAINE has joined the engineering staff of the National Engineering Co., Chicago. He formerly was president of the McIlvaine Foundry Conveyors Co.

♦

CHARLES F. BALL, formerly connected with Chain Belt Co., Milwaukee, has become connected with Joy Mfg. Co., Franklin, Penna., as director of engineering.

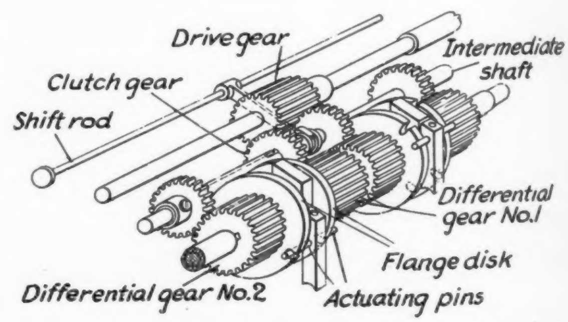
Noteworthy PATENTS

Depth of Tap Controlled

COUNTING device incorporated in a drilling and tapping machine accurately controls the depth to which a hole is tapped and automatically reverses the tap spindle. Assigned to Baush Machine Tool Co., the disclosed invention is adaptable wherever a predetermined number of revolutions of a shaft is desired before reversal such as in indexing or feeding mechanisms.

In the present application a driveshaft, geared directly to the tap spindle, carries a drive gear which transmits power to the counter mechanism. Mounted parallel to the driveshaft is an intermediate shaft which, at its midpoint, has a pinion in constant mesh with the drive gear. Splined to a sleeve on the intermediate shaft and capable of positive engagement with the pinion is a clutch gear which continuously engages the first differential gear. Fixed to the same sleeve is a second pinion which permanently engages the second differential gear.

Each of the differential gears has affixed to it a flanged disk on the periphery of which is formed



Differential gear train actuates limit switch after a predetermined number of rotations of the tap, reversing the spindle

an axial slot. Actuating pins engage these slots when they are in register. Upon engagement, a switch is tripped, operating the reversing mechanism. To be operative, the first differential gear has one less tooth than the second.

In operation, the shift rod is drawn to the left, disengaging the clutch gear. The second differential gear is then rotated so that its slot is out of phase with that of the other by an amount corresponding to the desired number of revolutions as indicated by a dial on the housing. The clutch is then engaged.

Hence, as soon as the tap has cut the predeter-

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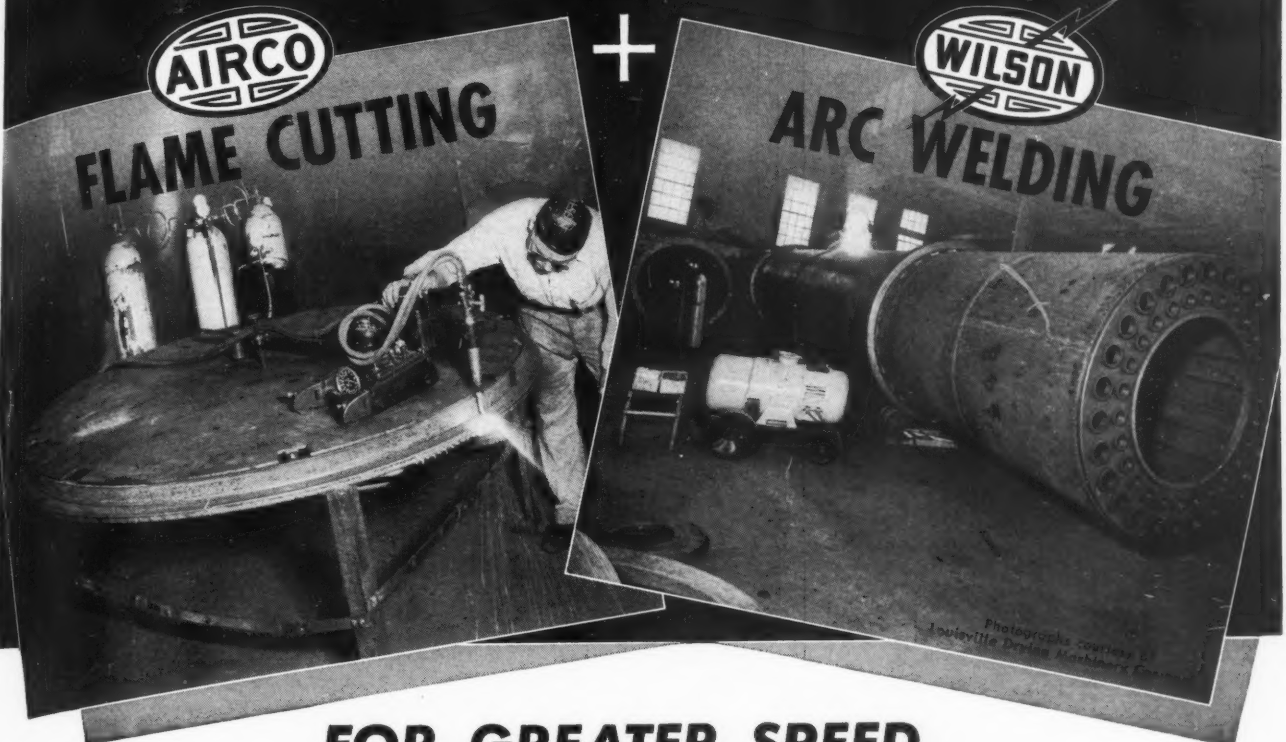

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Here's a splendid example: This large rotary dryer is being economically fabricated from 3/8" stainless clad plate speedily flame cut with an Airco Radiagraph using Airco 99.5% pure oxygen and Airco acetylene. Longitudinal and circumferential joints are made by using 1/2" x 4" steel butt straps plug welded and fillet welded to the inside shell plates. High quality welds and fast welding speeds are obtained by using the Wilson Hornet and Airco No. 81 Shielded Arc Electrode. A stainless steel inside seal weld completes the joint.

Flame cutting provides a quick, economical means of shaping steel into any desired contour whether a single unit or hundreds of identical pieces are required. Smooth, accurate cuts are assured. Parts are light, strong, tough. Arc welding unites flame cut parts into a single homogenous structure. Simplicity of design and greater efficiency of structure are important benefits; fabrication time and overall costs are lowered.

A most valuable advantage that goes with Airco products is the assistance of Airco's nation-wide Applied Engineering Department, available to every customer for instruction in the correct application of the welding and cutting processes and equipment to specific work, and for the solution of special problems. Be sure to ask for complete information about this service — a service worthy of the name.

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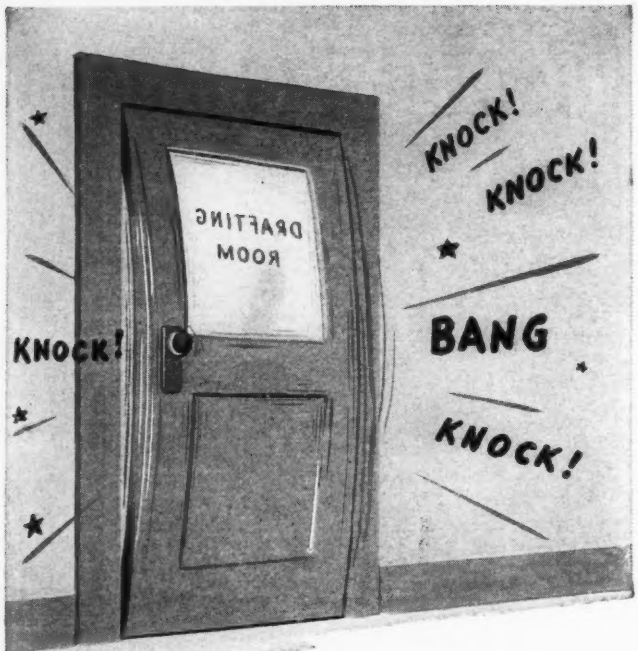


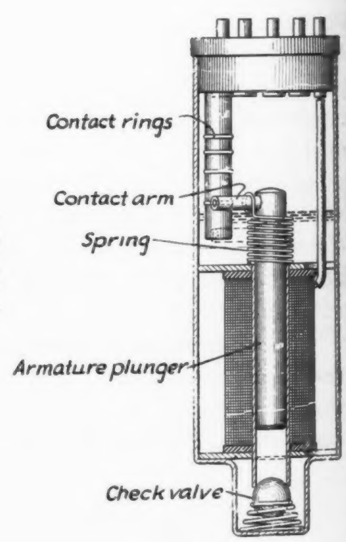
mined number of threads the slots come into register, trip the switch and start the reversing mechanism. The part of the counting device shown on the right is similar to that discussed above and serves to stop the machine when the tap has been completely extracted from the hole.

Controls Circuits in Sequence

OUTSTANDING attribute of the electric timer or sequence controller is its simplicity and low cost. Sequence control, as described in a patent assigned to A. E. Rittenhouse Co., is obtained by a built-in oil dashpot which directly affects the movement of a solenoid armature plunger carrying the contact arm.

A cylindrical tube affording a free fit with the armature forms, at its lower end, a seat for a spring-loaded ball check valve. This valve, when in the closed position, is intentionally designed to leak moderately. Contact rings of any desired number are mounted on a post attached to the head of the control. Each ring is connected to its own binding post. Contact with these rings is accomplished by means of a contact arm mounted on the armature at right angles to its longitudinal axis. Constant contact of the arm with the ring post is assured by a combination torsion and compression spring. In addition to providing contact, the spring also serves to raise the armature to its starting position when the solenoid is deenergized.

In operation the solenoid circuit is completed, moving the armature to the position shown in the illustration. When the solenoid is de-energized the check valve closes, and the slight leakage induces



Oil-operated plunger dashpot delays armature recovery, providing controlled sequence of operations

the plunger to rise slowly under the impetus of the spring. Spacing of the contact rings on the post determines the time delay for each interval. Whereas contact is momentary, it is apparent that the control can, if used with holding relays, serve as a pilot to a main power circuit. As shown and de-

(*Concluded on Page* 122)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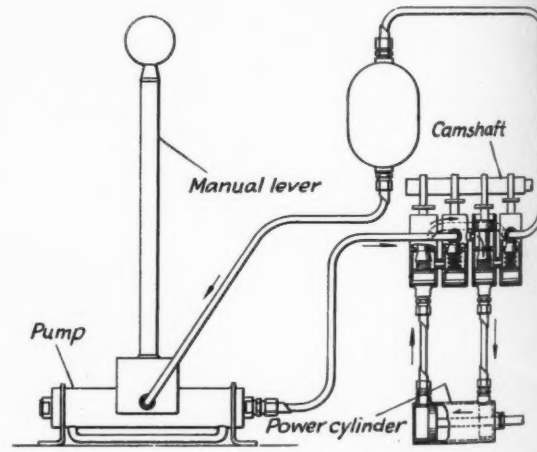
(*Concluded from Page* 116)

scribed, contact is also made in reverse order when the solenoid is energized. However, the time may be too short to actuate auxiliary relays. If this is not the case, the starter switch which initiates the sequence can serve as a control to hold open the pilot circuits during the return of the plunger.

Reverses Fluid Flow

INTENDED primarily for the remote control of retractable landing gear and other adjustable mechanisms on aircraft for which hydraulic power is used, the circuit illustrated possesses several noteworthy features. In the patent, assigned to Curtiss-Wright Corp., provision is made, by means of a bank of four external valves, to reverse the direction of flow to the power cylinder without increasing the number of lines to the pump or requiring the pump to reverse its displacement direction.

Each valve of the bank is operated separately by its own cam mounted on a single camshaft. The cams are so arranged that two valves are open at a time. As illustrated, the first and third from the left are open and the flow of fluid is as indicated by the arrows. When the second and fourth are open the piston of the power cylinder travels in the opposite direction.



Bank of four spring-loaded poppet valves permits reversal of power cylinder by means of a part turn of camshaft

The two middle valves, which may be called the upstream valves, are interconnected on the pressure side. The other two, the downstream valves, are each connected to one of the upstream valves on the cylinder side in addition to being interconnected on their downstream side.

Overloading of the camshaft bearings is avoided by making the stems of the poppet valves of sufficient diameter where they pass through the backing so that pump discharge pressure is compensated for in large measure. However, working cylinder pressure is effective against the valve heads and must be overcome in operating the valves.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Materials for Design of Pressure Vessels

(*Continued from Page* 48)

c. The diameter is less than $120t - 50$, where t = thickness in inches.

Fig. 5 shows graphically when such vessels must be stress relieved, as evaluated from part (c) of the foregoing. A vessel so constructed is recognized as having an efficiency of welded longitudinal seam of 80 per cent of that of the solid plate in computing its wall thickness.

Limitations Not To Be Exceeded

For service conditions of minor severity the rules of paragraph U-70 (Class III) may be applied, provided the following limitations are not exceeded:

a. The vessel must not be used for the storage of lethal gases or liquids (in addition to those listed under paragraph U-69, this class of construction does not permit the storage of ammonia, chlorine, natural or manufactured gas, propane or butane, even though these are not considered lethal)
b. The vessel must not contain liquids materially exceeding their boiling temperature at atmospheric pressure
c. The maximum working pressure cannot exceed 200 pounds per square inch
d. The maximum gas temperature cannot exceed 250 degrees Fahr.
e. The shell thickness cannot exceed ⅝-inch.

Under this class of construction the welders must be qualified in order to demonstrate their ability. No X-ray of the welds is required nor the stress relieving of the completed vessel. The maximum allowable working pressure of the vessel is calculated on the basis of a maximum unit working stress ($S \times E$) of 8000 pounds per square inch.

In the design of a pressure vessel, certain optimum conditions necessarily prevail. Obviously, the smaller the diameter of the vessel the thinner will be its wall thickness, based on the formula

$$t = \frac{P \times R}{S \times E} \qquad (1)$$

where

t = shell thickness, inches
P = maximum working pressure, pounds per square inch
R = inside radius of cylindrical shell, inches
S = allowable working stress, pounds per square inch = 1/5 of the minimum tensile strength
E = efficiency of longitudinal joint.

On the other hand, a small diameter will result in a long length of vessel, necessitating more circumferential joints. The number and size of openings and connections also may be a governing factor. Large diameters not only result in excessive wall thickness, but often run afoul of railroad clearances, size of annealing furnaces or size of shell and head plates normally rolled at the steel mill.

Having determined the diameter and length of the cylindrical vessel, the type of head is the next consideration. The semi-ellipsoidal form of head,

in which half the minor axis (or the depth of the head) is at least equal to one-quarter of the inside diameter of the head, has been proved to be strongest. It is the shape which a plate tends to take when subjected to pressure on the concave side. Its minimum thickness is computed by formula 1 where E = 1 (when made of a single sheet).

For moderate pressures, a two-radius head may be employed, with the radius of dish equal to or less than the diameter of the head joined to a "knuckle" radius equal to three times the thickness of the head, but in no case less than 6 per cent of the diameter of the head. Thickness of head is

$$t = \frac{8.33 \times P \times L}{2 \times TS} \quad \ldots\ldots (2)$$

where

t = thickness of plate, inches
P = maximum working pressure, pounds per square inch
L = radius to which head is dished, measured on the concave side of the head, inches
TS = tensile strength of the plate, pounds per square inch.

This type of head (known as A. S. M. E. flanged and dished head) is considerably thicker than the shell to which it is joined, requires a smaller flat circle plate and is not as deep as the semi-ellipsoidal type of head. When either of these forms of heads is subject to pressure on the convex side (for vacuum service or for steam-jacketed heads), the maximum allowable working pressure permitted is 60 per cent of that for heads of the same dimensions with the pressure on the concave side. In other words, formulas 1 and 2 are modified by the insertion of the factor .6 in the denominator. *Fig.* 4 shows dimensions and capacities of these heads.

For small diameters of cylindrical vessels, where a dished head may be difficult to form and costly, or when a butt weld may be impracticable, a flat head may be welded to the shell from the outside only, as in *Fig.* 7. The thickness of such a head is computed by the formula

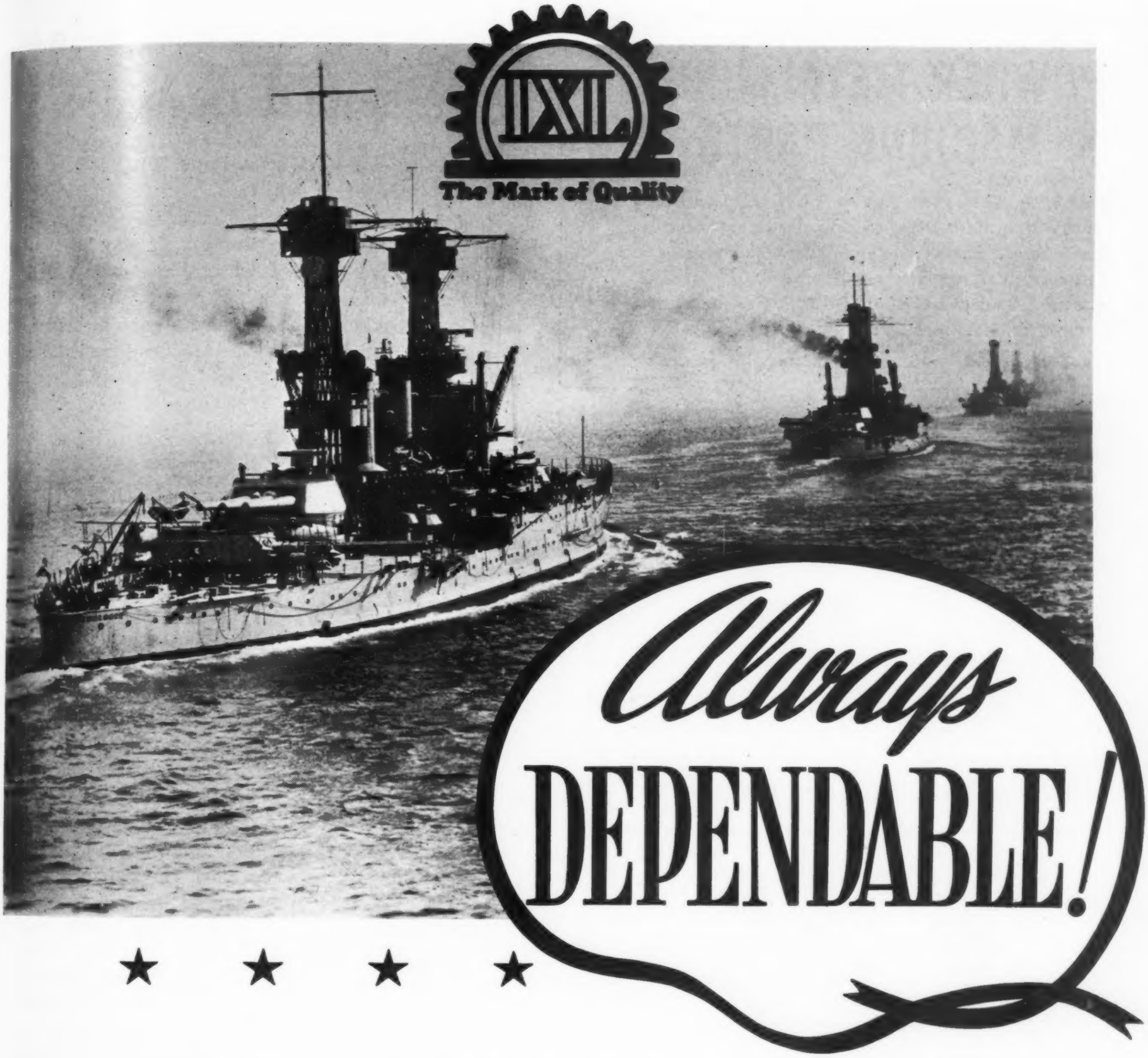
$$t = d\sqrt{\frac{C \times P}{S}} \quad \ldots\ldots (3)$$

where

t = minimum thickness of plate, inches
d = inside diameter of vessel, inches
C = constant = .5
P = maximum working pressure, pounds per square inch
S = maximum allowable unit working stress, pounds per square inch.

Thus far we have discussed vessels subjected to internal pressures. The engineer may also be confronted by a situation where the pressures are external to the vessel. Under such conditions the thickness of the shell may be computed from the accompanying chart, *Fig.* 6. Dished heads under these conditions are described in the section on heads.

In this discussion are presented the most important phases of the A. S. M. E. code, enough to give the engineer a broad view of the general problem of pressure vessel design, and a starting point toward the solution of that problem.



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GEAR AND MACHINE CORPORATION

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Theory of Elasticity in Practical Design

(*Continued from Page* 70)

Stress distribution from random concentrated loads, *Fig.* 73, is apt to be complex. It may be computed at any point R by combining simple radial systems from each load point O_1, O_2, etc., with a hydrostatic tension. The radial systems, as before, are obtained from the solution for the semi-infinite plate and are given by

$$S_\rho = \frac{-2P \cos\theta}{\pi b \rho}$$

The hydrostatic tension is given by

$$S = \frac{1}{\pi b d^2} \Sigma Pa \qquad (129)$$

Fig. 74 is a light field photoelastic picture of the disk with three loads.

Stress produced in the end of a chisel, in the tooth of a milling cutter or in many other cutting tools has its theoretical expression in *Fig.* 75 which has a concentrated load applied to the end of an infinite wedge. The solution of this shape usually will afford only a rough approximation to the stresses, as neither St. Venant's principle nor the theoretical requirements are easily satisfied. There will be no difficulty in finding sections sufficiently far away from the loaded area of the tip to permit the assumption of a concentrated load. But, from the nature of the shape itself, it usually will be impossible to get relatively far from the base. The base will be large and, the longer the wedge, the larger the base. The actual stress therefore, will depend to a considerable extent upon the method of fastening the base of the wedge to the main body of the part, or on the method of loading the base.

Although the practical problem cannot be exactly solved, a sufficient approximation usually can be made to assist materially in the design. At least some idea of the stress distribution will be gained. If a closer solution is desired recourse must be had to photoelastic analysis.

Solution for the general case, *Fig.* 75, is obtained from the stress function

$$\phi(\rho, \theta) = \rho\theta(A_1 \sin\theta + A_2 \cos\theta)$$

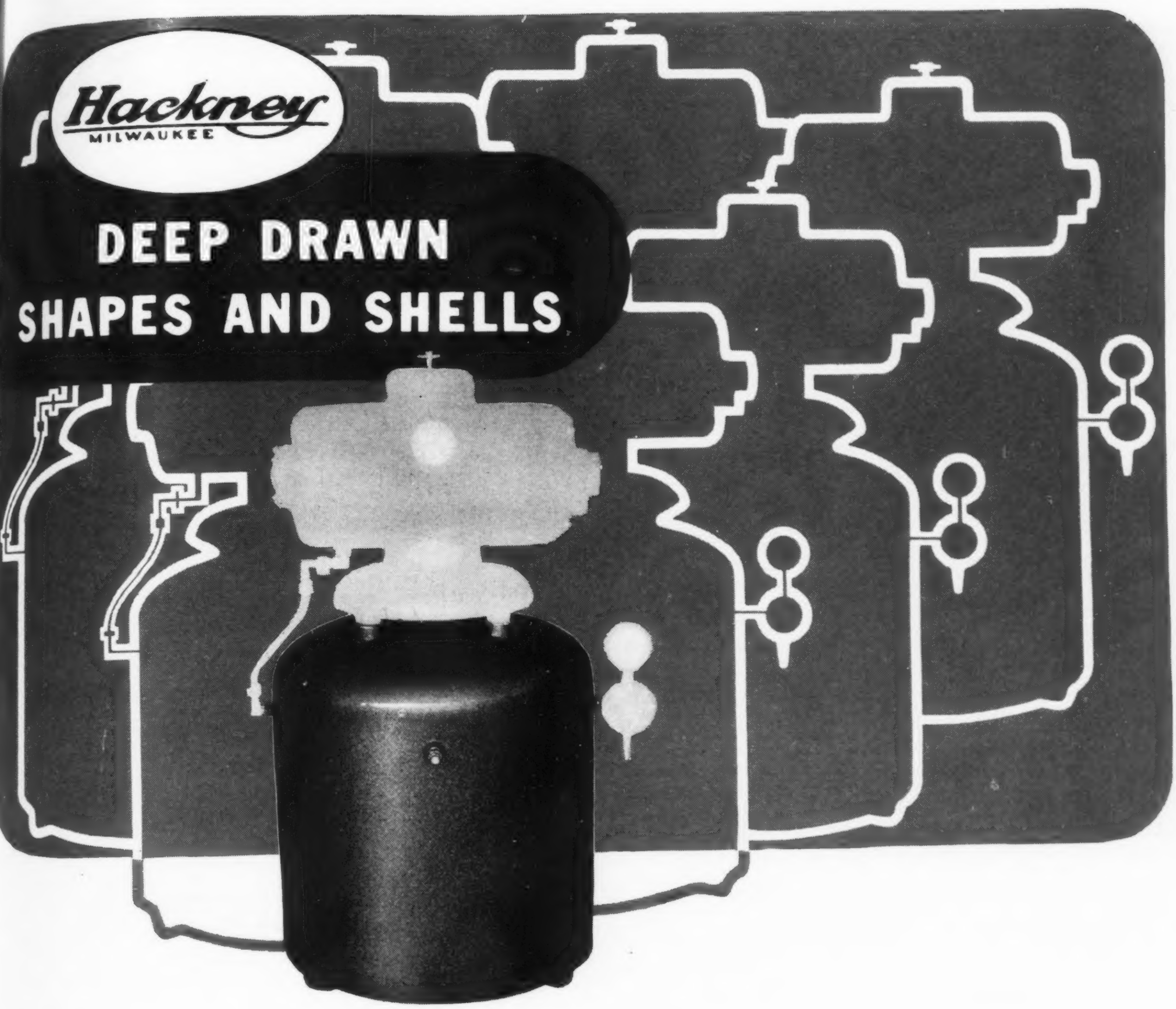
which checks Equation 66, establishing compatibility. The stresses from this function are, from Equotations 63, 64 and 65

$$S_\rho = \frac{1}{\rho}\frac{\partial\phi}{\partial\rho} + \frac{1}{\rho^2}\frac{\partial^2\phi}{\partial\theta^2} = \frac{2}{\rho}\left[A_1 \cos\theta - A_2 \sin\theta\right]$$

$$S_\theta = \frac{\partial^2\phi}{\partial\rho^2} = 0$$

$$v_{\rho\theta} = -\frac{\partial}{\partial\rho}\left(\frac{1}{\rho}\frac{\partial\phi}{\partial\theta}\right) = 0$$

Principal directions are radial and tangential. S_θ being zero the radial boundaries are free. S_ρ



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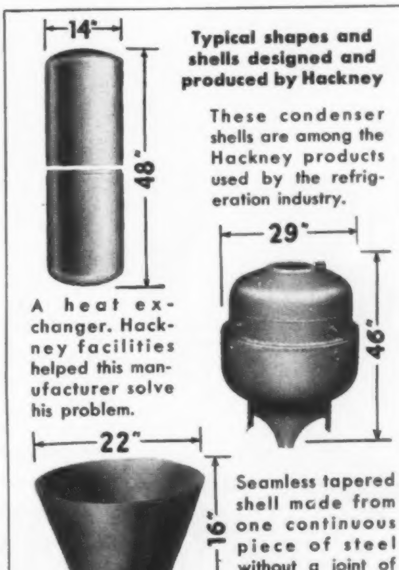

Typical shapes and shells designed and produced by Hackney

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A heat exchanger. Hackney facilities helped this manufacturer solve his problem.

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approaches zero as ρ increases without limit. Similarly, as in the semi-infinite plate, the constants are determined by the remaining condition that the summation of the forces on an arc of radius r, *Fig.* 75, must balance P. That is

$$\int_{\alpha-\beta/2}^{\alpha+\beta/2} S_\rho \, b r d\theta \cos\theta = -P$$

$$\int_{\alpha-\beta/2}^{\alpha+\beta/2} S_\rho \, b r d\theta \sin\theta = 0$$

Substituting the value of S_ρ and integrating between the limits gives two simultaneous equations to solve for the constants. Substituting the value of the constants back in the equation for S_ρ gives finally for the radial stress

$$S_\rho = \frac{-2P[\beta\cos\theta - \sin\beta\cos(2\alpha-\theta)]}{(\beta^2 - \sin^2\beta)\,b\rho} \quad \text{......(130)}$$

When $\beta = \pi$, the wedge becomes the semi-infinite plate and Equation 130 reduces to Equation 105 of the preceding section. Thus the plate problem is a special case of the wedge.

When $\alpha = 0$ the load is parallel to the axis, giving the symmetrical loading of *Fig.* 76. Equation 130 then reduces to[2]

$$S_\rho = \frac{-2P\cos\theta}{(\beta + \sin\beta)\,b\rho} \quad \text{......(131}$$

The solution is the same as that of the semi-infinite plate, Equations 105 and 106, except that π is replaced by $\beta + \sin\beta$. The discussion given holds in large part and will not be repeated. The radial boundaries will have a stress tangential to the surface, obtained by substituting $\theta = \beta/2$ in Equation 131.

Similarly to the plate problem, the stress along the circle of diameter g, *Fig.* 76 is constant and is given by

$$S_\rho = \frac{-2P}{(\beta + \sin\beta)\,bg} \quad \text{......(132)}$$

Q is equal to S_ρ and is given by Equations 131 and 132. The fringe pattern will be a series of arcs of circles centered on the load line and passing through the origin. A qualitative view of the pattern may be obtained from *Fig.* 50 by trimming it to the shape of the wedge.

When $\alpha = \pi/2$, P will be at right angles to the wedge axis. This solution is of particular interest because it gives the effect of the "tangential" component of the load. The stress reduces to

$$S_\rho = Q = \frac{-2P\cos\theta}{(\beta - \sin\beta)\,b\rho} \quad \text{......(133)}$$

The fringe pattern will be as sketched in *Fig.* 77, except at the tip where the fringes have been omitted due to the closeness of the spacing. Note

[2] Solution of the wedge with parallel load by means of the stress function of the semi-infinite plate is credited to J. H. Mitchell, 1902.

that the Q stress (twice maximum shear) is a maximum along the face, and increases at a greater rate as the point of the wedge is approached. This high surface shear stress means that fatigue failure is apt to manifest itself as a minute pitting of the surface, somewhat as has frequently been reported for gear teeth[3].

Appearance of the part is likely to lead to the conclusion that failure occurred from abrasive wear, whereas the exceeding of the fatigue strength of the material at the surface is a possibility. This could mean that some parts, gripping dog teeth for example, should preferably be made with a thin strong case than of a through hardening material. This matter is being carefully explored by the writer's company in the case of certain high production parts. Preliminary tests indicate a life at least as good as the production design, with an estimated production cost of little more than one-half!

[3] See "How To Reduce Surface Fatigue" by S. Way, M. D., March, 1939.

Designing Shafting

(*Continued from Page* 60)

In many problems the stresses $S_y' = S_y'' = 0$ and Equations 21 and 22 simplify to

$$S_w = \frac{1}{p}\left[\sqrt{(S_x')^2+3(S_s')^2} - (1-p)\sqrt{(S_x'')^2+3(S_s'')^2}\right] \quad (21a)$$

and

$$S_w = \frac{1}{2p}\Big[\sqrt{(S_x')^2+3(S_s')^2} - \sqrt{(S_x'')^2+3(S_s'')^2} + \sqrt{(S_x')^2+3(S_s')^2 - 2[(S_x')^2+3(S_s')^2]^{1/2}[(S_x'')^2+3(S_s'')^2]^{1/2} + [(S_x'')^2+3(S_s'')^2](1+4p)^2}\Big] \quad (22b)$$

Working stress values for fluctuating combined stresses are given by Equations 21 and 22 considering the fatigue properties of the material. Equations 10 give the values of the working stresses, based on the A.S.M.E. code, for which a fatigue load factor is applied. It is of interest to compare this code with the proposals. This will be done in the following example of a rotating shaft subject to bending and twisting moments.

EXAMPLE 3—ROTATING SHAFT SUBJECT TO BENDING AND TWISTING MOMENTS: If the shaft shown in *Fig.* 1 is rotating about the longitudinal axis, the diameter, according to the A.S.M.E. code for gradually applied loads, is obtained by replacing the moment M in the first of Equations 7 and 7a by 1.5M. This gives values of the diameter of

$$d_1 = (2.17)\left(\frac{T}{S_w}\right)^{1/3}\left(\sqrt{2.25R^2+1}\right)^{1/3} \quad \ldots\ldots\ldots (23a)$$

$$d_1 = (2.17)\left(\frac{M}{S_w}\right)^{1/3}\left(\sqrt{2.25+(R')^2}\right)^{1/3} \quad \ldots\ldots (23b)$$

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The values of the diameter given by Equations 23 include all possible load ratios as in the static case. The variation in the diameter by the A.S.M.E. code for different load ratios is shown in *Fig.* 9.

Allows Smaller Diameter

In considering the fatigue strength of the material using the failure condition defined by line AC, *Fig.* 6, it is only necessary in determining the diameter to replace the values of the stress components in Equation 21 by their values in terms of the loads. These stress components are

$$S_x' = \pm \frac{32M}{\pi d^3}, \quad S_x'' = 0, \quad S_y' = S_y'' = 0$$

$$S_s' = S_s'' = \frac{16T}{\pi d^3} \quad \text{(24)}$$

Placing these stress values from Equation 24 in Equation 21, an expression can be determined for the diameter required. This value is

$$d_2 = \left(\frac{16T}{\pi p S_w}\right)^{1/3} \left(\sqrt{4R^2+3} - (1-p)\sqrt{3}\right)^{1/3} \quad \text{(25)}$$

where $R = M/T$.

All possible load ratios can be included by using values of R from 0 to 1 and $1/R = R'$ from 0 to 1. These load ratios are considered in plotting the variation in diameter shown in *Fig.* 9. The values plotted in this design chart are for $p = S_e/S_{yp} = .8$. This is a reasonable value for medium carbon steels. The variation in the diameter with change in value of the material constant p is shown in *Fig.* 10. An examination of the design chart of *Fig.* 9 shows that the proposal gives a diameter less than the A.S.M.E. code—the difference being of appreciable magnitude in some cases.

If the failure condition defined by line AB in *Fig.* 6 is used, the stress values from Equation 24 are substituted in Equation 22 instead of 21. The diameter in this case is given by the value of d_2 in the equation

$$d_2 = \left(\frac{16T}{\pi p S_w}\right)^{1/3} \left(\sqrt{R^2+.75} - .87 + \sqrt{R^2+1.5+3p^2-(3R^2+2.25)^{1/2}}\right)^{1/3} \quad \text{(26)}$$

A design chart similar to that of *Fig.* 9 is shown in *Fig.* 11. This chart also shows that there may be considerable difference between the proposal and the A.S.M.E. code. This example is illustrative of several combinations of fatigue loadings which are encountered in design. In each case design charts can be constructed as shown in the examples given which will aid the designer in simplifying the calculations usually necessary.

There are many other factors than the combined stress effect which must be considered in the design of shafting. One main effect which should be mentioned is the influence of keyways on the strength of shafts. The keyway produces a stress concentration factor[8] which increases the stress at times as much as 75 per cent.

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MACHINE DESIGN

BOOK DEPARTMENT

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Restrictions Challenge Machine Designers

(Continued from Page 63)

rubber is being replaced by plastics, both thermosetting and thermoplastic.

As regards neoprene, plant expansion and continued research and development has increased production from an annual rate of a mere 3 million pounds in 1939 to 20 million pounds to be realized this year. In addition, plant facilities are being further expanded to produce an additional 22 million pounds annually. As an irreplaceable material, this ever increasing supply from native raw materials is distinctly encouraging.

STEEL—With the exception of high alloy steels which are scarce by reason of their alloying constituents the steel situation is excellent. By the fall of this year present production facilities will be so adjusted to demand that all defense industry will be supplied, plus an estimated 75 per cent of civilian needs.

Difficulties with steel supply are more likely to arise out of shortage of specific rolled sections or sheet and strip of specific gages than from any lack of material.

Reduce Tin in Bearings

TIN—This is one metal of which we have none within our borders, being completely dependent on the East Indies and South America. Arrangements have been consummated with Bolivia to smelt tin in this country, eliminating transshipment to Britain. Under present conditions this, along with our considerable stocks, should be adequate if due precautions are taken in its conservation.

In many instances, high-tin babbitt in bearings can be replaced satisfactorily with babbitt using less tin. Similarly low-tin babbitts may be replaced by lead babbitts. On the other hand, bearing substitutions in direction of higher load capacities may be met by leaded bronze or sintered bronze.

As in the case of steel, bronze shortages are felt more in rolled sheet than in basic material shortage. In replacing sheet for bearing uses, bronze has been melted and "puddled" on steel backings for automotive piston pin bearings.

TUNGSTEN—In large measure the present level of machine tool production is dependent upon sintered carbide tools of which a considerable proportion is tungsten. In this case also, increasing price is accelerating the development of domestic deposits which promise, shortly, to be adequate. Heretofore practically our entire supply has been obtained from the interior of China.

For many uses of tungsten, molybdenum is a capable substitute. In this regard it is interesting to note the number of alloying elements which may

(Concluded on Page 142)

(*Concluded from Page* 136)

be replaced adequately by molybdenum in the event of necessity. This comparatively new industrial element is rapidly achieving a place of eminence in industry.

VANADIUM—Little concern need be felt regarding this metal. Visible supplies are adequate for our requirements.

ZINC—The zinc situation is another which may shortly be expected to relax. New plants for its production are being built in Wisconsin by Anaconda. Similar plants are under construction by Chase. In addition a new plant in Utah will be soon in production, reclaiming zinc from slag.

Satisfactory substitutes for zinc die castings are lead-antimony alloys although this material does not take as good a finish. Other alternatives are pressed steel, plated cast iron, plastics either plain or sprayed with bronze and plated.

Facilitating Engineering Functions

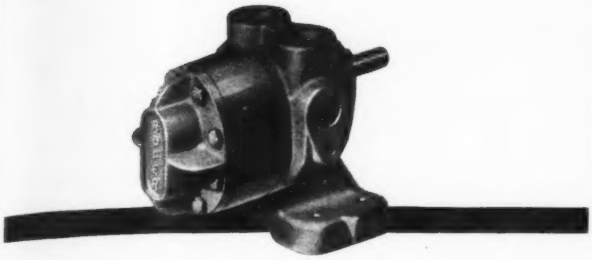
Problems of shortages in skilled and experienced engineering manpower are, if anything, more critical than scarcities of materials. This is true both for the reason that there can be no adequate substitute and also because the time element for development is generally greater.

Recognizing this condition, at least as regards designers of tools, jigs and fixtures, the suggestion has been made that a group of such specialists be organized into a sort of flying squadron. Such a group could then descend on a plant, literally like a "flock of locusts," tool up the place and go on to the next. It is reasonably certain that the job done in this manner would be better, quicker and cheaper.

Detailing is another bottleneck in engineering departments. In many plants shortages of detailers require the designer to make his own detail drawings, resulting in excessive loss in efficiency. The tempo of work in developing an original design is inevitably slow. It is next to impossible to avoid carrying over this same tempo to the execution of details.

If the designer had only to develop assemblies and important sub-assemblies his time would be immeasurably conserved. These drawings could then be turned over to an independent organization of draftsmen specializing in detailing, as distinct from the "flying squadron" of tool designers. For checking, the details could be returned to the original designer.

Fortunately, we already have the nucleus of such an organization in the many student training courses being conducted throughout the country by schools and private industry. In a well-intentioned effort to assist their designers, many companies in the defense industries have so filled their organization with untrained engineers, that the designers, in supervising the work and correcting mistakes, are actually hindered. It would be well, at least "for the duration", if engineering department functions could be streamlined in the manner of a production line so that each specialized ability could be used to maximum efficiency.



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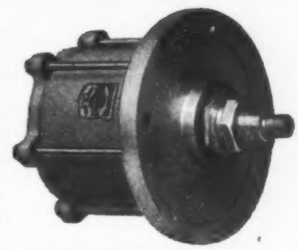

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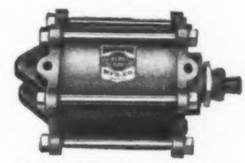

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"As a result of research and invention, fifteen million American men and women are working in jobs that did not exist in 1900."—E. O. Shreve

"Prime emphasis will be placed on military products for five or more years. The sooner we accept the fact, for purposes of planning, that we face a long period of enormous production for defense, with a consequent shortage, rationing and allocation of strategic materials, the more quickly will nondefense industries adjust themselves to the new conditions and undertake the great task of maintaining maximum production for civilian needs without interference with defense output."—Philip D. Reed.

"The defense program brings the whole subject of standards into sharp focus. A manufacturer who takes a contract for a product which differs from his regular line of production is faced with the necessity of making many changes in the equipment and operation of his plant. His problems are much like those of an automobile manufacturer in retooling for a new model. These problems are basically problems of standardization."—Howard Coonley and P. G. Agnew.

Meetings and Expositions

Aug. 25-29—
National Association of Power Engineers Inc. National Power Show and Mechanical exposition, and fortieth annual meeting will be held at the Fifth Regimental Armory, Baltimore. Fred W. Raven, Room 1717, 176 West Adams street, Chicago, is secretary.

Sept. 8-12—
American Chemical society. Semiannual meeting to be held at the Chalfonte Haddon hall, Atlantic City, N. J. Dr. Charles I. Parsons, Mills building, Washington, is secretary.

Sept. 23-26—
Association of Iron and Steel Engineers. Annual meeting to be held at Statler hotel, Cleveland. Brent Wiley, 1010 Empire building, Pittsburgh, is managing director.

Sept. 28-Oct. 2—
American Mining congress. Annual metal mining convention and exposition to be held at Fairmont hotel, San Francisco. Julian D. Conover, 309 Munsey building, Washington, is secretary.

Oct. 6-10—
National Restaurant association. Annual meeting and exhibition to be held at the National Restaurant Mart, Chicago. Frank J. Wiffler, 666 Lake Shore drive, Chicago, is secretary.

Oct. 13-14—
Steel Founders' Society of America. Fall meeting to be held at The Homestead, Hot Springs, Va. Raymond L. Collier, 920 Midland building, Cleveland, is secretary.

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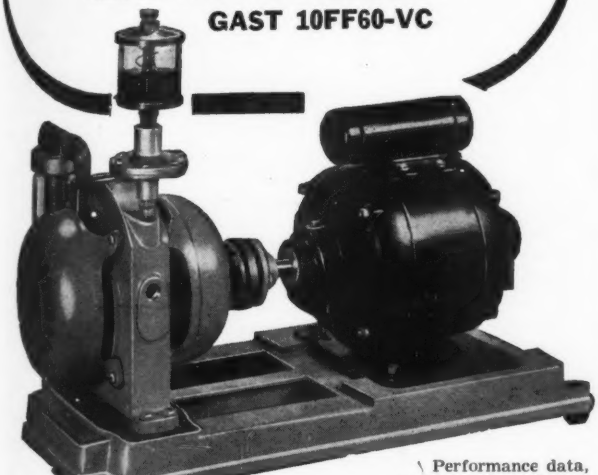

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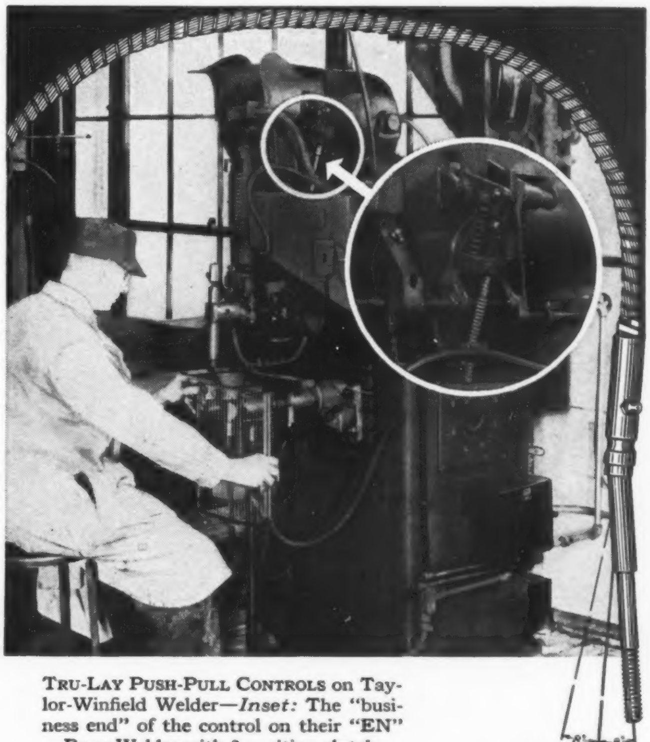

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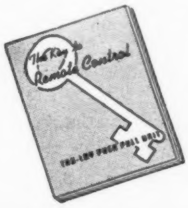

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Business and Sales Briefs

FOR several years an associate professor in mechanical engineering at the A. & M. College of Texas, James A. Trail has joined the Hyatt Bearings division, General Motors Sales Corp., and will cover the southwest territory, traveling out of Dallas. A. J. Swisler, who formerly handled this territory, will work out of the western division headquarters of the company in Chicago.

♦

In its newest plant addition, the Michigan Tool Co. is devoting all operations to production of cone-drive gearing to keep pace with the growing demand for this form of gearing. The plant, a two-story combination construction, is located adjacent to the company's main plant on East McNichols road, Detroit. The former Cone Worm Gear division will now be known as Cone Drive division.

♦

Appointment of J. Earl Romer as district manager of the Cleveland office of Bliss & Laughlin Inc. has been announced recently. Mr. Romer who succeeds A. W. Schultz, resigned, has been associated with the company for five years, beginning in the Cincinnati sales area. During the past three and a half years he was assigned to the sales staff at Cleveland.

♦

Formerly manager of sales in industrial control section of General Electric Co., G. R. Prout has been appointed manager of the industrial control division. R. S. Glenn will assist Mr. Prout as sales manager.

♦

In charge of western New York territory of Ampco Metal Inc., Milwaukee, for some time, Sherman Barnes has established new headquarters at 699 Potomac avenue, Buffalo.

♦

Despite "bombs and blitzes" the International Meehanite Research institute held its annual meeting in London recently, according to word reaching the Pittsburgh office of Meehanite Metal Corp. Representatives from the 19 Meehanite foundries in Great Britain attended.

♦

As a result of an increased demand for bonded rubber mountings, particularly in the aviation industry, the Lord Mfg. Co. has moved its western office to larger quarters at 245 East Olive avenue, Burbank, Calif. Duff Dean is western district manager. The company has also announced the opening of a new district office at 4937 Laclede avenue, St. Louis, Mo., with George Harrington in charge. He will represent the company in southern Illinois, Indiana, Missouri, Kansas and the southwest.

♦

The fifth unit of the Aluminum Co. of America was placed in operation in Vancouver, Wash., recently, increasing the capacity of the plant to more than 150,000,000 pounds of aluminum annually. This figure is significant when considering that the total production of all aluminum in the United States did not exceed 130,000,000 pounds per year in the last world war, and that the entire industry in the United States did not produce 150,000,000 pounds a year until 1924. Yet the new Vancouver works will account for only a fifth of the metal produced by Aluminum Co. of America by midsummer of 1942.